Machinery for Refrigeration

MACHINERY

FOR

REFRIGERATION

BEING

SUNDRY OBSERVATIONS WITH REGARD TO THE PRINCIPAL APPLIANCES EMPLOYED IN ICE MAKING AND REFRIGERATION, AND UPON THE LAWS RELATING TO THE EXPANSION AND COMPRESSION OF GASES. PRINCIPALLY FROM AN AUSTRALIAN STANDPOINT

BY

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CHICAGO
H. S. RICH & CO.
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INTRODUCTION.

Among the many marvelous strides which the nineteenth century has witnessed in connection with the arts and sciences, those which have been made in the commercial production of cold hold a very important place. The evolution of artificial refrigeration from the theoretical and experimental to the practical stage hardly dates back forty years, and its present vast proportions have only been approached during the last quarter of the century. The production of ice was probably the chief incentive to the work of the early inventors, but there can be no doubt that the preservation and transportation of food products, and the requirements of industries connected with cold storage, are largely responsible for the remarkable development of artificial refrigeration in later days.

The mechanical processes carried out in an ordinary refrigerating establishment are, when compared with many others in which machinery is employed, exceedingly simple, but they are dependent upon principles which are not so easy to comprehend; and perhaps no branch of engineering has been less understood in the past, by those who use machinery, than that which is connected with ice making and refrigeration. The only books, at one time, which threw any light on the subject, dealt with it simply from the thermodynamic aspect, and for their due comprehension required the reader to be a mathematician rather than a refrigerating engineer. There are now many trade catalogues, issued by makers of refrigerating machinery, which give useful information, both

as to theory and practice, and some are of exceeding merit in the scope and accuracy of the information which they furnish.

The establishment of a journal like *Ice and Refrigeration* not only evidences the importance of the refrigeration business, but it forms a means of communication between refrigerating engineers all over the world, and disseminates the knowledge of every improvement to the five quarters of the globe—five, because Australia, where it is largely read, is not included in the orthodox four. Apart from this, the proprietors of that journal have published their "Compend," which to-day is the rule of faith to thousands of persons who have charge of, or are interested in, refrigerating machinery. More recently the same publishers have issued another book by "The Boy" (Mr. Skinkle), and there are a number of English works dealing with the history and progress of refrigeration, all supplying information under one or more of the many aspects which the subject presents.

The author commenced his connection with refrigerating machinery in the year 1858, and with the exception of the years 1884 and 1885, when he studied its progress and improvement in the United States and Europe, he has been in Australia ever since. He should thus look at American and European rival refrigerating machines with an unprejudiced eye. His first writings on the subject were penned in the endeavor to do justice to some of the Australian pioneers in refrigeration, such as Harrison, Mort and Nicolle, whose important labors seemed to be ignored in American and European works. Other papers by him have since then been read before the Royal Society of New South Wales, and the Southern Ice Exchange of the United States, in connection with the same subjects, and have been so kindly received outside the colony that he has now been induced to attempt to write a whole book.

In the following pages the reader must not expect to

find anything new from the theoretical side. First principles never alter, and there are many books available for those who wish to dive into the thermodynamic principles involved in the operation of the machines employed for artificial refrigeration, but it is believed that a great many matters relating to the construction and practical working of such machinery, as well as to the distinctive characteristics of different refrigerating systems, are now presented, either in a new shape, or for the first time. To the average ice or cold storage man who wants to produce the greatest amount of cold, with the least primary investment of capital, the smallest cost of maintenance, and the lowest working expenses, this little work may possibly be of some service; and if the author should at any future time learn that brother engineers—like himself, more practical than literary—have been helped by what follows to a fuller understanding of the requirements and possibilities of a modern refrigerating plant, it will give him the satisfaction of knowing that his efforts in this connection have not been altogether misapplied.

NORMAN SELFE.

SYDNEY, N. S. W., AUSTRALIA.

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HISTORICAL.

Over 300 years are supposed to have elapsed since it was first discovered that artificial cold is produced by the chemical action which takes place when certain salts are dissolved, but it is not known how far back the system of making ice has been practiced which is still in use in India, where shallow trays of porous material are filled with water and exposed to the night air, so that the heat may be abstracted by the natural evaporation which takes place. The use of frigorific mixtures for the abstraction of heat (many forms of which are still set out in works on chemistry) was known as far back as the year 1607, and the most common combination, that of ice and salt (which is said to have been used by Fahrenheit in 1762, when he placed the freezing point of water at 32° as the limit of negative temperature), is still in every day use for such purposes as ice cream freezing.

The production of cold by what may be termed mechanical means (that is by the use of a refrigerating machine as distinguished from chemical action) is of much more recent date. Dr. Cullen is said to have made a machine for evaporating water under a vacuum in 1755, and Lavoisier experimented with ether in France, but the next important steps appear to come well into the present century. In the year 1810 Leslie experimented with a machine using sulphuric acid and water. In 1824 a machine was patented by Vallance, who probably got his idea from the evaporative system so long used in India. Under this patent, dry air was circulated over shallow trays of water when evaporation took place and heat was abstracted.

In 1858 Mr. George Bevan Sloper patented a similar system in New South Wales.* Under this invention the water to be frozen was contained in canvas bags, so that the

* N. S. W. L. R., No. 14, 1858.

whole surfaces of such vessels were exposed to the evaporative effect of the surrounding air as well as the surface of the water itself. The machine to work this process was designed by the author to carry out the ideas of the patentee just forty-one years ago. It was constructed in Sydney by Messrs. P. N. Russell & Co., then the leading engineering firm in Australia, and tried in Margaret street, Sydney. No commercial success, however, did or could attend any such system of producing artificial cold, owing to the excessive

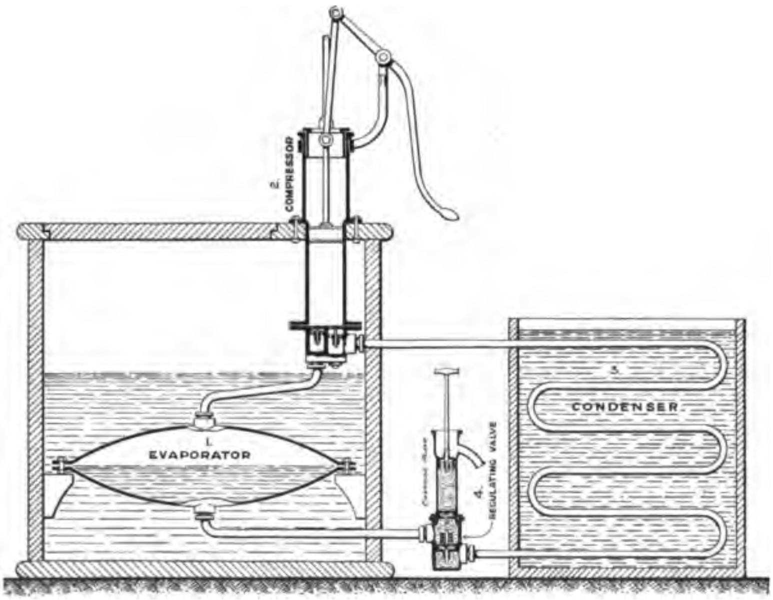


FIG. 1.—JACOB PERKINS' ICE MACHINE, PATENTED IN 1834.

amount of power required to produce a given result; and in this particular case, as the air delivered into the chamber under partial vacuum was not made to perform work on its way from the atmosphere, it did not part with the equivalent heat beforehand, and therefore did not reduce the temperature of the water, as it might have been made to do, had the knowledge of thermodynamic laws at that time been as widely extended as it is now.

In 1834 Hagen used the volatile spirit of caoutchouc, and in the same year Jacob Perkins, of London, constructed what appears to have been the first ice making machine which

really worked successfully with a volatile liquid. In this machine of Perkins' ether was vaporized and expanded under the reduced pressure maintained by the suction of a pump; and the heat required for such vaporization was abstracted from the substance to be cooled. The resulting vapor was then compressed by the same pump into a vessel cooled by water, until under the influence of the increased pressure the vapor parting with heat to the cooling water again condensed to a liquid, and this liquefied medium was then ready to be evaporated and expanded over again.

Fig. 1 is taken from Jacob Perkins' English patent, No. 6,662, of August, 1834, and shows clearly that his invention included the four principal features still in use in all modern compression systems, viz.: The evaporator (1), the compressor (2), the condenser (3), and the expansion or regulating valve (4) between the condenser and the evaporator.

Although his machine was the forerunner of all the compression systems of the present day, Perkins does not appear to have had any more success in introducing it for commercial uses than Vallance had. Dr. Gorrie, in 1845, seems to have taken the steps which led to the invention of the cold air machine, with which the names of Windhausen, Bell, Coleman, Haslam, Lightfoot, Hall, Giffard and others are associated, and which were the first class of machines that were successful in carrying meat from Australia to Europe. In 1850 Carré invented the ammonia absorption process. Between the years 1850 and 1860, Professor Twining in America, and Mr. James Harrison, of Geelong, in Australia, devoted themselves to the improvement of Perkins' ether machine, probably without either inventor knowing what the other was doing, as there was not much communication between the two countries in those days. Twining is said to have had a machine at work between 1855 and 1857 in the state of Ohio, and Harrison, in the year 1855, was at work in Victoria when he actually produced ice with fish frozen in the block. In the year 1859 the Harrison machines were introduced into New South Wales, and manufactured by Messrs. P. N. Russell & Co.; the author, who at that time was in the drawing office of the firm, was connected with this work from its initiation.

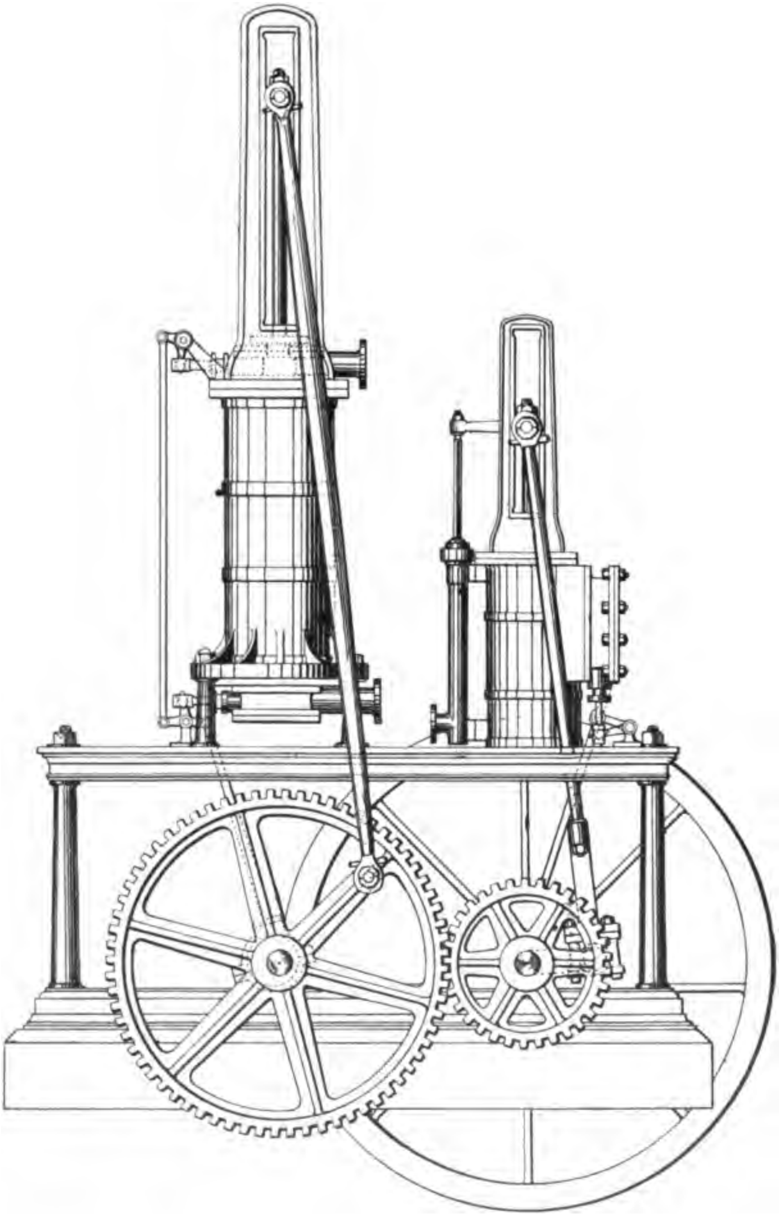


FIG. 2.—HARRISON'S ETHER MACHINE—TABLE PATTERN. 1859.

The original drawing of these machines is now in his possession, and Fig. 2 is a reproduction from it. As will be seen from the figure, they were made as a double-table engine with four slide valves to the ether pump, a separate inlet and outlet valve on the top and bottom covers being worked by cams and an eccentric. One of them, when completed, was set to work at the rear of the Royal hotel, George street, Sydney, and supplied ice to a regular list of customers; another and similar machine was sent to Melbourne.

In the same year (1860) P. N. Russell & Co. made more Harrison machines to a horizontal design prepared by the author, who was then their chief draftsman. These worked for many years in New South Wales and Victoria, and were illustrated in *Ice and Refrigeration* for February, 1895. A large double-cylinder machine, designed by the author also in 1861, is shown by Fig. 3, on the following page.

Messrs. Siebe, of London, had introduced the Harrison into England about this time, and it is generally admitted in both America and England that the very first ice machine ever adopted successfully for manufacturing purposes was one of Harrison's Australian ether machines, applied to the extraction of paraffine from shale-oil in 1861. The *Engineer* for April 12, 1861, has an illustration of Harrison's machine as made by Siebe.

Dr. Kirk invented a sort of regenerative air machine in 1862, which was also used for the cooling of paraffine oil in Scotland. From the years 1861 to 1870 Mr. E. D. Nicolle, of Sydney, worked at the development of the ammonia absorption system, first introduced into France by Carré, the latter years in conjunction with the late Mr. T. S. Mort. In 1863–64 he made a pump to compress anhydrous ammonia to the liquid condition, which proved tight at 30 atmospheres. He, however, considered the absorption system as the more economical in fuel, and his machines at Darlinghurst quite supplanted the Harrison ether machine in George street. Many thousands of pounds were spent by Mr. Mort in experiments not only with the ordinary absorption system (many practical improvements in which were patented), but on a compressed air system, L. R., No. 181, of 1868, on an absorption or "affinity" system operated by a pump, L. R., 216, of 1869

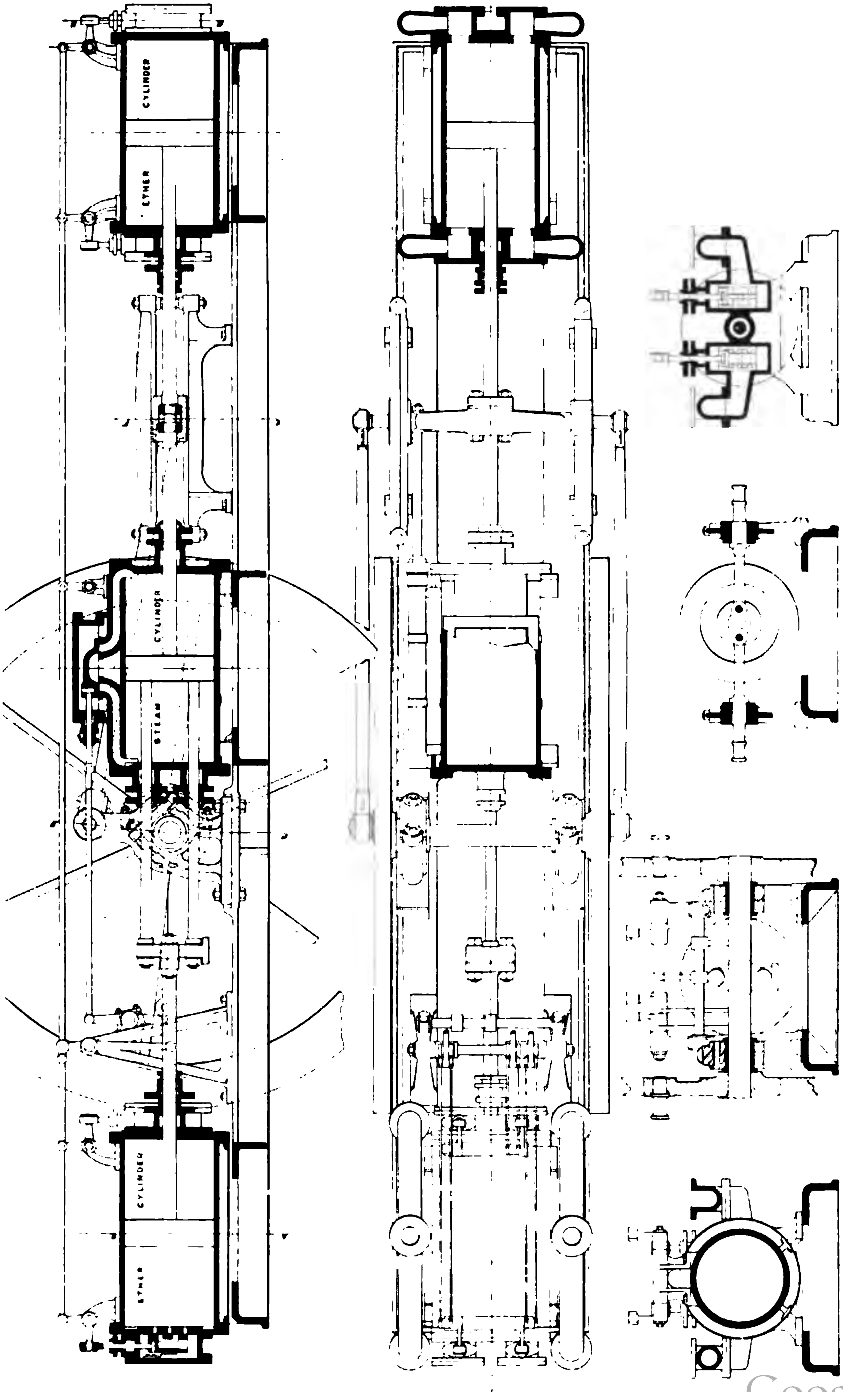


FIG. 3.—HARRISON'S DUPLEX (ETHER) ICE MACHINE. DESIGNED BY THE AUTHOR IN 1861.

(see *Ice and Refrigeration*, April, 1899, page 298), and also on a system of using nitrate of ammonia, which was fitted up in the ship *Northam*, all under the direction of Mr. Nicolle.

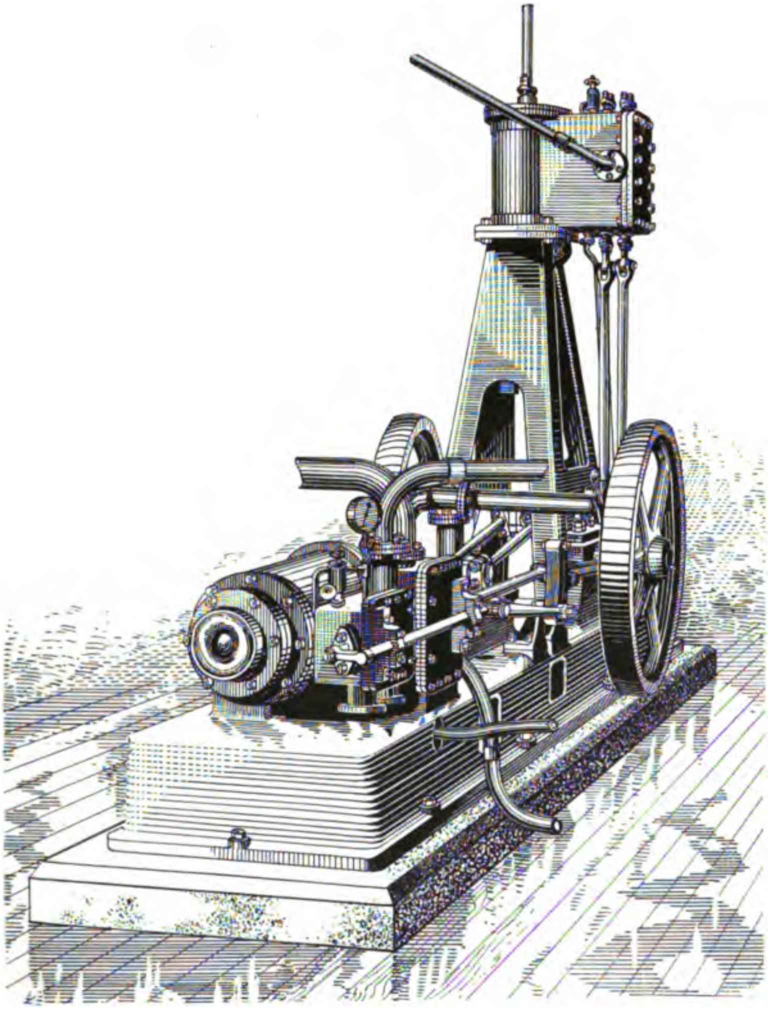


FIG. 4.—COLD AIR MACHINE, WITH COMPOUND EXPANSION.

The first practical compression machine designed in New South Wales, for the use of anhydrous ammonia as a refrigerating medium, was patented by the author (No. 887, of 1880), and was called the "Colonial Freezing Machine." It

embodied many devices which are now in general use. (See Fig. 68.) In 1885 the late Mr. W. G. Lock, chief engineer to the Fresh Food and Ice Co., of Sydney, patented a compound compressor for ammonia (L. R., No. 1,729). This consisted of two single-acting high and low pressure pumps, side by side, very similar to the machines now being made by the York Manufacturing Co., of York, Pa., U. S. A. In 1881 the author designed the compressed air machine illustrated by Fig. 4 for a bacon curing business in a country district where there were only untrained men to work it and water was very scarce. Special condensers were adopted, and the water was used over and over again, less the waste by evaporation.

This machine has worked successfully up to the present time and, though recently supplemented by an ammonia machine, will still deliver air at 50° below zero F.

As will be noted from the illustration, the expansion is compounded and, although the high and low pressure valves are worked from one eccentric, they are connected to separate blocks in a double link motion so as to allow for different grades of expansion being adjusted to each. By this means the temperature of the intermediate chamber in the sole plate can be so regulated as to secure the deposition of moisture there, without affecting the cut-off and expansion in the other cylinder.

Great numbers of patents have since been issued in New South Wales to local engineers for compressors of more or less originality, and for other details of refrigerating machinery, and it must not be forgotten that Mr. J. D. Postle, by his New South Wales patent, No. 180, of August 24, 1868, was one of the first persons in the world to understand and patent the use of an expansion cylinder in a cold air machine. By this means some of the heat held by the air is converted into work, and a lower temperature is produced. It will be seen from this short account that New South Wales has in the past done a large share of the work by which the refrigerating machinery of the world has been brought to its present perfection.

It is probable that in the United States the development of the ice machine has been due more to its use in the brewery and to the national taste for iced water than to other

applications, and that in New South Wales the idea of freezing food products for export, first suggested in 1860 by the late Augustus Morris—when he offered to contribute £1,000 toward the experiment of sending frozen meat to England—was the main factor which induced the late Mr. T. S. Mort to devote his energies, and probably a quarter of a million pounds sterling, toward the economic production of artificial cold. For more particulars as to Mr. Mort's great work the author would refer those interested to an article he contributed to *Ice and Refrigeration* for August, 1895.

CHAPTER II.

ON HEAT AND COLD.

The words "hot" and "cold" taken by themselves convey no definite ideas of temperature; they are merely relative terms. To any one coming in from a snow storm, water at 60° would feel warm, but to a person in a steamer's stokehold, at 120° or more, the very same water might feel refreshingly cool. If sentient beings exist under the atmosphere of the sun, then the temperature of molten iron and gold, or the great heat—to us—of 2,000°, may be, relatively, colder to them than we find ice cream in this world. We really know nothing about a limit to possible heat, that is temperature in the direction of its increase, although we may admit that there is a condition of things in the universe where all matter exists as vapor. We have, however, through modern research, the knowledge that there is a limit to the disappearance of heat, and that a condition is possible below which there could be no further reduction of temperature; and, as the production of cold means the abstraction of heat, at this theoretical foundation or zero point all heat must have disappeared and absolute cold be reached.

From the thermometric base thus set up the properties and effects of heat can be measured and compared, and absolute cold being the bottom of the scale, all degrees of *temperature* are simply relative. It is better not to make a practice of speaking of degrees of *heat*, because the word heat is applied in several different ways; and it will be well before going further to refer to four separate expressions inseparable from our subject, in which the word is used, viz., sensible heat, latent heat, heat unit and specific heat.

SENSIBLE HEAT.

Sensible heat or temperature is that heat or "hotness"

which is apparent to our senses and which we can measure with a thermometer. There are three differently scaled instruments principally used for this purpose. The French or Centigrade thermometer is considered by many the best for laboratory work and scientific research; the Reaumur is used in Germany and in some breweries; but in ice making and refrigerating establishments where the English language is spoken, the Fahrenheit thermometer is the one in universal use; therefore, whenever temperatures are referred to in these pages, unless specially excepted, they apply to Fahrenheit's scale. The zero mark or 0 on this instrument is placed 32° below the freezing point of water; such freezing point is marked 32°. The boiling point of water at sea level is 212°, and the absolute zero of temperature is 460° below zero, or 492° below the freezing point of water. What the intensity of this cold amounts to will perhaps be better understood and appreciated when it is considered that just as much as ice is colder than molten solder, so much is absolute zero colder than melting ice.

HEAT UNIT.

A heat unit is a standard of measurement by which is expressed the capacity of a given weight of any body to absorb and retain heat or energy, under a given increase of sensible heat or temperature. As water possesses a greater capacity for heat than almost any other known body, its properties have been adopted for such standard. In scientific circles the French unit or "calorie" is used, that being the amount of heat required to raise one kilogramme of water 1° (from 4° to 5°) Centigrade; but the British thermal unit, written B. T. U., which represents the heat required to raise one pound of water 1° (from 39° to 40°) F., is the universally recognized standard of heat measurements among ice men in America, England and Australia.

SPECIFIC HEAT.

The specific heat of any body or substance is the capacity for heat which any given weight of that body possesses as compared with an equal weight of the standard, water. As the specific heat of water is expressed by unity, the spe-

cific heat of any other body is a fraction, which represents the proportion of a British thermal unit that is required to raise the temperature of one pound of any such body 1° F. The specific heat of different bodies is found to vary slightly with change of temperature, but, although this amount has to be allowed for in calculations of scientific accuracy, it is of no importance in the practical work of refrigeration. The specific heat of gases, however, varies very much under the two different conditions of their pressure in the one and their volume in the other case, being affected by the addition or abstraction of heat. This will be referred to more fully when dealing with the compression and expansion of gases.

LATENT HEAT.

The term latent heat is applied to the heat which is taken up by any body without causing a change of temperature, when it passes from the solid to the liquid state, or from liquid to vapor; in the former case it is termed the latent heat of liquefaction or fusion, and in the latter case the latent heat of vaporization. When either ice or a metal is melted from the solid state at its respective melting point, or water at 212° is converted into steam at 212°, then heat is taken up without a change of temperature or sensible heat, and such heat is said to be latent; and, similarly, when the reverse process takes place this latent heat is again given out.

ILLUSTRATIONS OF HEAT TERMS.

From the table of specific heats which follows it will be seen that the specific heat of ice is .504. The latent heat of liquefaction of water is 142 B. T. U., and the latent heat of its vaporization is 966 B. T. U. From this we are able to see that it will not suffice to abstract 212 units from steam at 212° to bring it down to ice at 0°, but that 1,304.5 units must be removed—as the quantity of measurement of thermal units—by which one pound of water differs as regards its storage of heat under the two conditions.

Thus, to reduce:

A pound	of steam	at 212° to	water	at 212°	represents	966.	B. T. U.
"	of water	at 212° to	"	at 32°	"	180.	"
"	of "	at 32° to	ice	at 32°	"	142.4	"
"	of ice	at 32° to	"	at 0°	"	16.1	"
					Total....	1,304.5	B. T. U.

The specific heat of iron is only between .11 and .13 (say only one-eighth part) that of water; and thus rubber bags or metal cylinders filled with hot water are used instead of hot metal, when it is required to store heat for such purposes as personal comfort. In old fashioned tea-urns it was customary to use an iron heater to keep the water warm, but owing to the low specific heat of iron, any given weight of that material cooling from 560° to 160°, or through 400°, would only give out the same amount of heat as an equal weight of water cooling to 160° from the boiling point.

THE HEAT TO BE ABSTRACTED IN THE WORK OF REFRIGERATION.

It is evident, therefore, that the work which has to be accomplished in cooling, chilling or freezing food, or other materials, and in reducing the temperature of the chambers in which they may be stored, is (apart from leakage and conduction) directly dependent upon the specific heat of the various substances involved, namely, those of which the chamber is constructed, of the air which it contains, and of the goods stored therein. The following table shows the specific heats, as generally accepted, of a few of the more common materials connected with the construction of refrigerators and the substances which are usually stored therein to be refrigerated, it being noted that the specific heat of food products is largely governed by the percentage of water which they contain.

Water	being 1.00	Marble	is .209 to .215
Ice	is .504	Air	is .238
Turpentine	is .467	Oil	is .310
Oak	is .570	Alcohol	is .659
Pine	is .650	Strong brine	is .700
Wrought iron	is .113 to .125	Vinegar	is .920
Cast iron	is .130	Cream	is .680
Tin	is .056	Milk	is .90
Zinc	is .095	Fat beef	is .60
Copper	is .095	Lean beef	is .77
Lead	is .031	Fat pork	is .51
Coal	is .241	Veal	is .70
Coke	is .203	Fish	is .70 to .85
Charcoal	is .241	Chicken	is .80
Brickwork (Rankine)	is .200	Fruit or vegetables	is .50 to .93
Glass	is .197	Eggs	is .76
Stone	is .270		

CHAPTER III.

THE PRACTICAL WORK OF ARTIFICIAL REFRIGERATION.

HOW CAN A MACHINE PRODUCE COLD?

Seeing that all machines work with more or less friction, and that the power thus lost reappears in another form of energy as heat—which is sensible and apparent—there is some excuse for the difficulty felt by the ordinary lay mind in comprehending the production of cold by machinery. It may be said at once that no combination of mechanism, even with unlimited power to drive it, could alone make ice from water; and that an ice machine is simply an instrument for dealing with some substance which operates as a medium in such a way that it (the medium) is enabled to take up heat from the body to be cooled and transfer it to another body.

Except under very special circumstances, which will be referred to later on, this heat is transferred to the water which is used for the purpose of condensation and goes to waste.

TWO CLASSES OF MEDIUM USED.

There are two distinct systems of mechanical refrigeration in use, both operating by means of a medium. Under the more simple system this medium is a permanent gas which is alternately compressed and expanded, but not liquefied under such compression. In actual practice atmospheric air is alone used for this purpose, and the machines are termed compressed air machines.

Under a more complex system of mechanical refrigeration a more volatile medium is employed, and in the operation of the machinery there is alternate liquefaction and volatiliza-

tion. Although many different media have been tried, each of which has some special quality to recommend it, the principal ones to which reference will here be made are sulphuric ether, sulphurous acid, ammonia and carbonic acid. In the system introduced by Carré a solution of ammonia in water is employed, the gas is driven off by the direct application of heat, and is again reabsorbed by the water after fulfilling its functions in the circuit of the apparatus; this is known as an "absorption system." Although there are many authorities who still advocate the advantages of the absorption process, the greater number of refrigerating engineers now adopt the compression system.

CHAPTER IV.

COLD AIR MACHINES.

These machines, coming under the first or simpler system already referred to, operate by virtue of the law that all mechanical work has a thermal equivalent. The diagram (Fig. 5) illustrates the action of such a machine in dealing with a pound weight of air. At atmospheric density, or 14.7

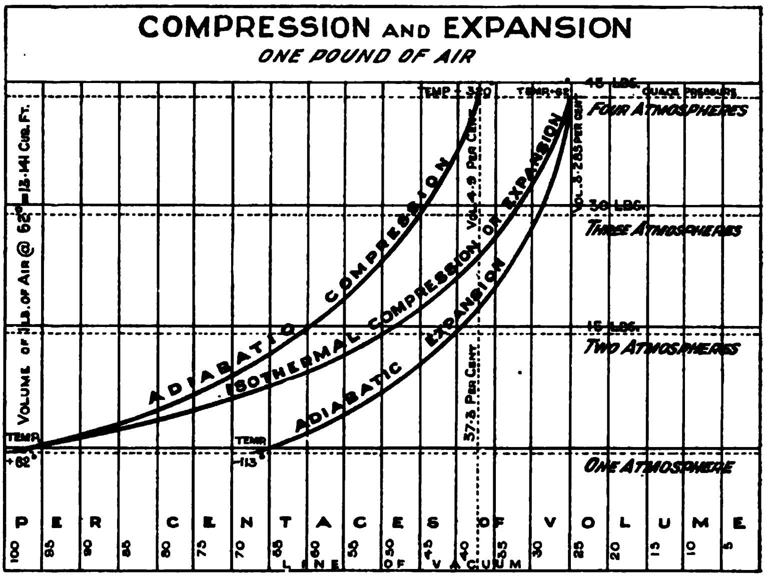


FIG. 5.—DIAGRAM ILLUSTRATING THEORY OF COLD AIR MACHINES.

pounds per square inch, and a temperature of 62°, one pound of air possesses the intrinsic energy due to its specific heat multiplied by its absolute temperature, *i. e.*, 62°+461°=523°; and it occupies a volume of 13.141 cubic feet, which is represented by the horizontal length of the diagram. If such a

volume of air is compressed to a density of four atmospheres, then between 47,000 and 48,000 foot-pounds of energy will be required to perform the work, and if we assume a frictionless piston and a non-conducting cylinder, the air will not follow Mariotte's law, and by an isothermal compression occupy one-fourth or 25 per cent of its original volume at the original temperature, but will rise to a temperature of 320°, and fill 37.3 per cent instead of 25 per cent of the original volume, the difference representing the work performed by the engine in the operation of compression.

Now while the medium is under this increased tension, which with cold air machines seldom exceeds five atmospheres, the compressed air may be passed through a condenser or cooler and have its temperature again brought to 62°, in which case the heat or energy expended upon it by the engine will be communicated to the condensing water, and for all practical purposes be lost. The air then only possesses the same intrinsic energy which it did before compression, but it is in a physical or mechanical condition which enables it to perform work by expanding again to atmospheric pressure. This expansion in practice is carried out in an engine similar to a steam engine, which assists the working of the whole refrigerating machine, and the final temperature of the air is found by simple proportion, thus:

As compressed absolute temperature before condensation is to compressed absolute temperature after condensation—that is, as—

$$461^\circ + 320^\circ = 781^\circ \text{ is to } 461^\circ + 62^\circ = 523^\circ$$

so is original temperature before compression to final temperature after expansion—that is—

$$461^\circ + 62^\circ = 523^\circ \text{ to } 348^\circ \text{ absolute} = -113^\circ$$

Or, to make a simple proportion sum of it—

$$781^\circ : 523^\circ :: 523^\circ : 348^\circ \text{ and } 348^\circ - 461^\circ = -113^\circ$$

In actual practice this theoretical low temperature is never reached, about —80° being the minimum, and —50° an ordinary temperature, the losses from friction and conduction being proportionately much less in larger machines, as would be supposed. The results with these machines are also much affected by the moisture in the air and other causes.

The shipment of chilled and frozen meat from Australia, the Argentine Republic, and the Cape, to England, led to a great number of ships being fitted with air machines, and several eminent firms in England made a specialty of their manufacture. Owing to the confined space available between decks on shipboard, these machines were often so placed as to make access to their working parts very difficult, and when they reached the largest size the whole transverse space available in the ship was filled. Fig. 6 represents one of the largest cold air machines as made by the Haslam Co., of Derby, England, and by its complication suggests that the engineer in charge would likely be glad to see it replaced by a modern ammonia machine. From an inspection of the illustration it will be seen that in this typical cold air machine

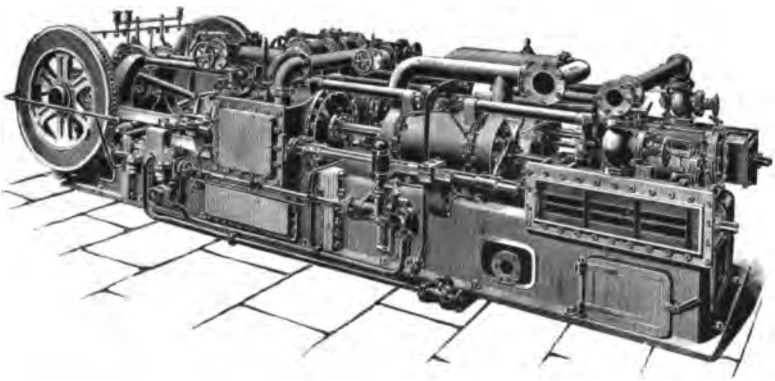


FIG. 6.—HASLAM COLD AIR MACHINE FOR SHIPBOARD.

(still greatly used) the high and low pressure cylinders of the compound steam engine are at the crank shaft end, the two compression cylinders in the middle, and the two expansion cylinders, with their special slide valves, at the tail end. The surface condenser, cooler and snow-box are all arranged in the bed plate, the latter below the expansion cylinders.

However complicated this machine may look, it is really a simple affair when compared with some cold air refrigerators that have been constructed. Fig. 7 represents the cylinders forming one side of a machine which is fitted with two compound tandem steam engines, two air compressors and two air expansion cylinders, all coupled to one crank shaft, and with steam condensers, air coolers and snow-

boxes in the sole plate. Such a machine was fitted in a ship to carry 80,000 carcasses of mutton from Australia, and it broke down at sea. If there is any ammonia man who thinks he has a hard row to hoe in looking after his own machine on land, perhaps he will just consider what is the first thing he would try to do if he was in charge of such a freezing machine as this, running eighty-five revolutions per minute with 160 pounds of steam pressure, the ship rolling heavily, the temperature in the holds going up, $80,000 worth of meat at stake, and his compressor pistons so much in trouble that it is certain they must be got out and be replaced by spare ones. In such a case, which actually occurred, the engineer in charge grappled successfully with it, and then had his sole

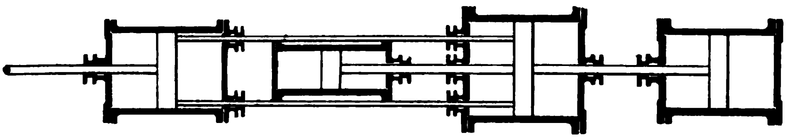


FIG. 7.—COMPOUND STEAM ENGINE, WITH COMPRESSION AND EXPANSION AIR CYLINDERS, FORMING ONE SIDE OF REFRIGERATING MACHINE FOR SHIPBOARD SERVICE.

Steam engines, 14" and 24" diameters. Compressor, 30". Expansion cylinder, 23" diameter. Stroke 30. Steam pressure, 160 lbs. Revolutions, 60 per minute.

plate break by the working of the ship and throw all out of line. This necessitated the transshipment of the whole cargo to another vessel in Sydney harbor after a voyage from Queensland had been completed.

GREAT POWER REQUIRED BY COMPRESSED AIR MACHINES.

Both in theory and in practice compressed air machines require very much more power for a given abstraction of heat, amounting to from four to six times as much as some other machines do. They are therefore rapidly going out of use except for special purposes. It is possible in compressing air to reach very high and low relative temperatures without much difficulty, and it occurred to the author, in the early days of refrigeration, that *some* of the heat or energy which is dissipated to the condensing water in these machines, *and which is equivalent to the whole amount of the engine power*, might be utilized by combining a compressed air refrigerator

with a modification of the Du Tremblay ether engine; and he took out a New South Wales patent in April, 1880 (No. 812), for a refrigerating machine which had an ether engine as well as a steam engine to supply the power.

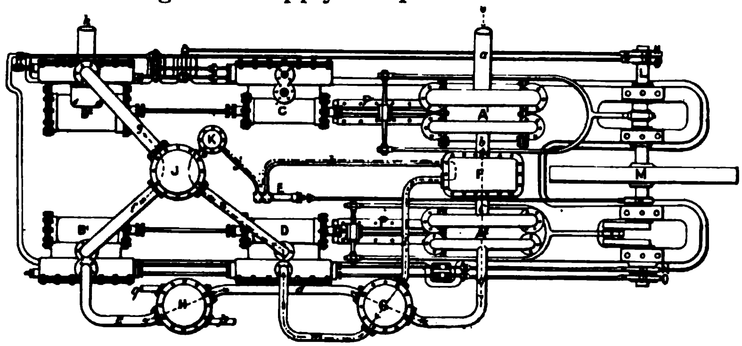


FIG. 8.—REFERENCE TO COMBINED AIR AND ETHER MACHINE.

A^1 Low pressure compression cylinder.

A^2 High " " "

B^1 High " expansion "

B^2 Low " " "

C Steam engine "

D Ether " "

E Ether pump.

F, G } Condensers or exchangers to transfer the heat of compressed air to vaporize ether.

H Ordinary surface condenser for water.

J Condenser or exchanger for liquefying ether vapors by expanded and cooled air.

K Receptacle for liquid ether.

L Crank shaft.

M Fly wheel.

N Slide valve eccentrics.

O Expansion valve eccentrics.

P Slipper guides.

a Inlet to first compressor from chill room (through desiccator or exchanger if used).

b Compressed air delivery to exchanger F.

c Inlet to second compressor from exchanger F.

d Compressed air delivery to exchanger G.

e Inlet to air expansion high pressure cylinder.

f Exhaust to exchanger J from high pressure cylinder.

g Inlet to low pressure air expansion cylinder from J.

h Expanded air exhaust to cold chamber.

j Suction pipe to ether pump E from vessel K.

k Delivery pipe from ether pump E to exchanger F.

l Pipe conveying heated ether from F to G for further heating.

m Pipe conveying ether vapor from G to ether engine F.

n Exhaust pipe of ether engine to condenser J.

o Pipe conveying liquid and condensed ether to vessel K.

p, *q* } Inlet and outlet pipes for condensing or circulating water to condenser H.

- - - - - - - - - - - Direction of air.

-•-•-•-•-•-•-•-•- Direction of ether.

In this machine the first heat was to be abstracted from the compressed air in a primary condenser or exchanger by means of ether sprays on the condenser tubes, and the vapor thus produced was to be utilized in the ether engine to assist the steam engine and reduce the steam power necessary for the work. The machine has never been made, and it is certain that in actual practice a very large percentage of the power thus saved would be required to overcome the extra friction resulting from the additional number of parts; still it appears absolutely certain that it is only by some such method of utilizing the heat which is now thrown away in the condensers of refrigerating machines that any great fuel economy in the future of artificial refrigeration is possible.

CHAPTER V.

THE USE OF A GAS WHICH LIQUEFIES UNDER PRESSURE.

In referring to the second or more complex system of mechanical refrigeration it was stated that a volatile medium such as ether, sulphurous acid, ammonia and carbonic acid was employed instead of a permanent gas, as in the air machines. Before considering the construction of machines used with these gases it will be well to consider some of the properties of the gases themselves.

PROPERTIES OF GASES MOST CONCERNED IN THE OPERATION OF REFRIGERATING MACHINES.

It is not many years since the liquefaction of carbonic acid and ammonia, now so much used in refrigerating machines, was confined to laboratory experiments; but since it has been understood that pressure and cold were the factors necessary to liquefy them, other gases, which it was for a long time considered impossible to deal with, including hydrogen, have been liquefied also. Condensed liquid oxygen is now sold as an ordinary commercial product, and air, after being liquefied by the gallon, has been frozen solid.

It is, therefore, possible that in the future there may be refrigerating machines operating with liquid air under the enormous pressure of, say, 2,000 pounds per square inch, with a primary condenser at, say, 90° temperature, and a secondary or tertiary condenser at —250°. At the present time (omitting methylic ether, used under the Tellier system in France) the principal media used in refrigeration machines are restricted to sulphuric ether, sulphur dioxide, ammonia and carbonic acid.

Now, as with the conversion of water into steam, all the substances just referred to require to take up heat to change

them from the liquid to the gaseous condition. It makes no difference to this property that the boiling point of three of them is below the ordinary temperature of the atmosphere, so that their normal condition at atmospheric pressure is the gaseous one. As with water and steam, the boiling point of these and other gases means the temperature at which such gases will liquefy, as well as that at which their liquids will pass again to the gaseous condition; in fact a temperature under which a given weight of the material may be either entirely liquid, entirely gaseous, or partly in one state and partly in the other, depending for its condition upon the number of heat units contained in or held by it; and such temperature, as with steam, depends upon the actual pressure to which it is subjected at the time. Conversely the pressure under which any gas can be liquefied depends upon its temperature.* At atmospheric pressure the boiling points of these four gases are as follows:

SULPHURIC ETHER.	SULPHUR DIOXIDE.	AMMONIA.	CARBONIC ACID.
+96°	+14°	—29°	—124°

For the practical purposes of artificial refrigeration the lowest temperature to which heated gases under pressure can be reduced is limited by the temperature of the water used for condensation.† This water may be as low as 45° or 50° in temperate countries, and in hot climates may exceed 90°.

The diagram, Fig. 9, shows in graphic form the vapor tensions of carbonic acid, ammonia, sulphurous acid, ether and water under the temperatures met with in practical work, or the boiling points of these media under widely varying conditions as to pressure. For instance, it will be seen that carbonic acid, which under atmospheric pressure will boil at 124° below zero, requires about 1,080 pounds per square inch to liquefy it at 96°, affording a great contrast to water, the boiling point of which at 14.7 pounds or one atmos-

*The critical temperature of a gas is that temperature above which no increase of pressure will produce liquefaction and the gas remains permanent.

†For experimental purposes to produce very low temperatures the condensed gas may be cooled by a second refrigeration, and a step-by-step process adopted for attaining the lowest extreme possible.

phere is 212°, and which requires the pressure reduced down to 0.089 of one pound per square inch (or a very high vacuum) to enable it to evaporate at 32°. Again, sulphuric ether, which boils at 96° under atmospheric pressure, must be attenuated to at least twelve pounds below atmospheric

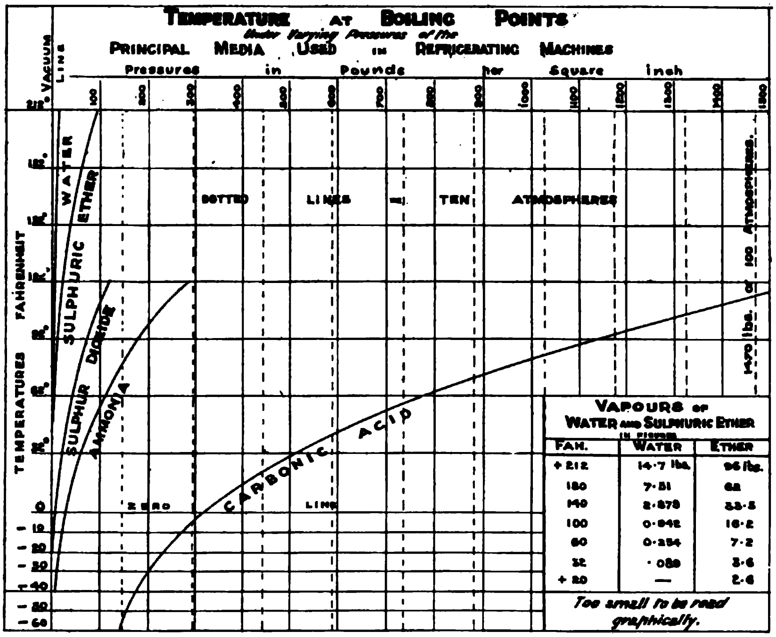


FIG. 9.—DIAGRAM ILLUSTRATING BOILING POINTS OF REFRIGERATING MEDIA.

pressure before it will evaporate at the freezing point of water. From these figures it will be noted that machines for making ice by the evaporation of either water or ether must work with a partial vacuum, their pumps exhausting their refrigerators to pressures below that of the atmosphere.

CHAPTER VI.

THE LATENT HEAT OF LIQUEFACTION IN ITS APPLICATION TO REFRIGERATION.

Although the temperature at which a volatile medium may be made to boil in the coils of a refrigerator has a very important bearing on the production of cold, as—other things being equal—the lower the degree of cold produced the greater the amount of heat that can be taken up, yet there is another property of these volatile substances which has a great deal to do with the results that can be attained by their use in a refrigerating machine, and that is their latent heat of liquefaction, or the number of heat units that any given weight of such medium will take up in passing from the liquid to the gaseous condition. To make the importance of this property clearer, we may suppose that a pound of one medium in evaporating will abstract heat enough to bring *two* pounds of the substance to be refrigerated down 100°, while one pound of another medium will, under similar conditions, lower the temperature of *ten* pounds of the same substance, but only by 50°; still the medium in the second case would, other things being equal, be two and a half times as efficient for the purpose of refrigeration, because it would, in its conversion into vapor, abstract two and a half times as many thermal units from its surroundings as that in the former one. Supposing liquid, such as wort in a brewery, is the substance to be cooled, then two pounds lowered 100° represents the abstraction of 200 thermal units, while ten pounds lowered 50° would be equivalent to 500 B. T. U.

Therefore, before we can ascertain the relative efficiency of two or more different media for abstracting heat in a refrigerator, we must ascertain their respective latent heats

of liquefaction under the conditions which accompany their practical application.

Fig. 10 is a diagram which shows in thermal units the latent heat of one pound of each of the four principal media

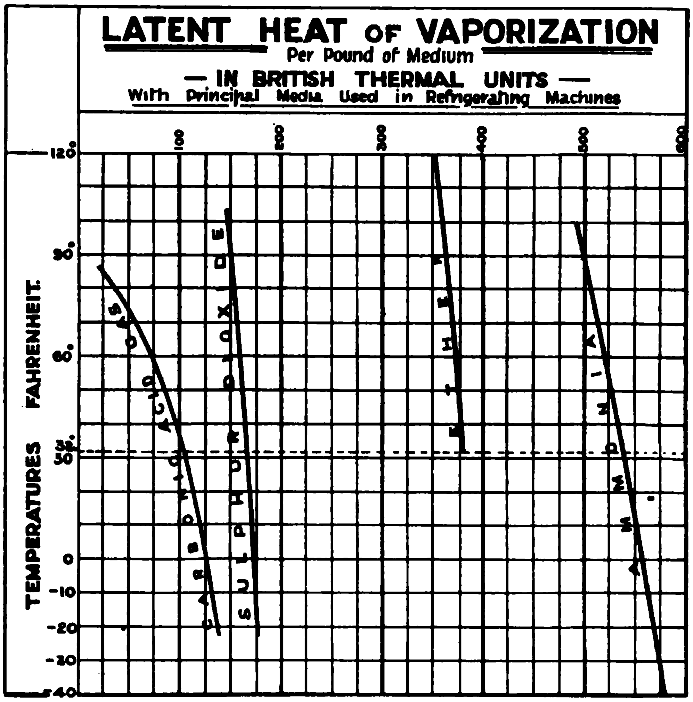


FIG. 10.—DIAGRAM SHOWING LATENT HEAT OF REFRIGERATING MEDIA.

before referred to, and under a range of temperature which covers their ordinary use for refrigerating purposes.

USELESS WORK PERFORMED IN THE REFRIGERATOR.

When a gas is liquefied under the influence of pressure (whether produced by a pump or through the direct application of heat), and the abstraction of heat by cooling it in a condenser, the resulting liquid is necessarily at a temperature something above that of the condensing water, and is still under the pressure at which it was condensed; but it is in a position to change its condition again directly that influence is removed. In actual practice the pressure is retained in the condenser, or liquid receiver, by an "expan-

sion" cock or "flash" valve, which regulates the passage of the liquid refrigerant into the coils of the refrigerator, releasing its pressure at the same time from that in the condenser to that of the refrigerator. Under these conditions the liquid on its release immediately boils and evaporates, or, in the words often used, *flashes* into vapor, hence the name of the valve. In so doing it abstracts heat from the metal of the coils and the air or liquid surrounding such coils; but it must be particularly noted that this gaseous medium has to be cooled down itself before it can cool the refrigerator to any given or required temperature, and that therefore a certain proportion of its actual cooling power is not effective for external refrigeration.

The amount of heat, or the number of thermal units, that is thus lost before any useful refrigeration is done is the product of the specific heat of such medium, multiplied by the number of degrees it is lowered in temperature. All this cooling power is absolutely lost so far as any useful effect is concerned, because the medium has to be heated up again by the expenditure of more energy at every circuit it makes through the machine.

THREE PROPERTIES OF A GAS CONCERNED IN FORMING AN EFFICIENT REFRIGERATING MEDIUM.

From the foregoing remarks it will be understood that the relative efficiency of different gases for refrigerating purposes is mainly dependent upon three properties possessed by them, and not upon any one special characteristic, and these are:

1. A low temperature of vaporization upon which depends the degree of cold that can be produced by such evaporation.

2. A high latent heat upon which depends the total number of heat units which will be abstracted by the evaporation of a given weight of the medium.

3. A low specific heat upon which depends the net percentage of the heat taken up by (2), or, in other words, the proportion of the gross amount of cold produced which can be actually utilized.

CHAPTER VII.

WHY AMMONIA IS SO LARGELY USED IN REFRIGERATING MACHINES.

Although ether, chloride of methyl and several other media have been used in refrigerating machines besides those already referred to, and some are still advocated under special conditions, yet ammonia is now used more than all the rest of them put together, experience having proved the many advantages it possesses. The principal reason why ammonia has supplanted the use of other liquids as the circulating medium in refrigerating machinery is because it has such a high latent heat of vaporization, being 555 B. T. U. at zero, against 123 for carbonic acid. That is to say, one pound of ammonia at zero in passing from the liquid to the gaseous condition would take up 555 thermal units, while the other liquids before referred to would take up less than a third and less than a fourth, respectively, of that amount.

There are some compensating advantages in the case of carbonic acid on account of its high specific gravity, which makes its heat of vaporization for a given volume very much greater than ammonia. The relative volumes at zero of equal weights of ammonia and carbonic acid are about 1 : 32.4, and thus the relative dimensions of the compressors for equal refrigerating effects are as $\frac{123.2 \times 32.4}{555}$ is to 1, which equals nearly 7.2 for ammonia to 1 for carbonic acid. This quality would be an advantage if all other things were equal, but carbonic acid reaches a critical condition at 88° F., and its efficiency rapidly falls off when the condensing water is above that temperature. Many carbonic acid machines have their refrigerator and condenser placed the one inside the other for the sake of compactness, as shown in Fig. 12, and although

not so intended by the makers, such device enables power to be expended to cool the condensing water if desired. Such an expedient is totally unnecessary with ammonia, and machines using that material often work with the condensing water at 90° or over without any great falling off in efficiency.

FIG. 11.—HALL'S CARBONIC ACID MACHINE.

On shipboard there have unfortunately been many incidents to create a prejudice against ammonia, which there is little doubt were largely the result of inferior workmanship and want of care, and as a consequence carbonic acid machines are now in great favor for vessels at sea. The saving of fuel in

such cases is very large when compared with the consumption by the old cold air machines, and therefore the still greater saving that could be effected with the use of ammonia plants under like conditions is not at present receiving much attention. No doubt when the suitability of ammonia machines for seagoing ships is better understood they will supplant the carbonic acid machine to some extent on account of their greater economy; but, owing to carbonic acid permitting the use of

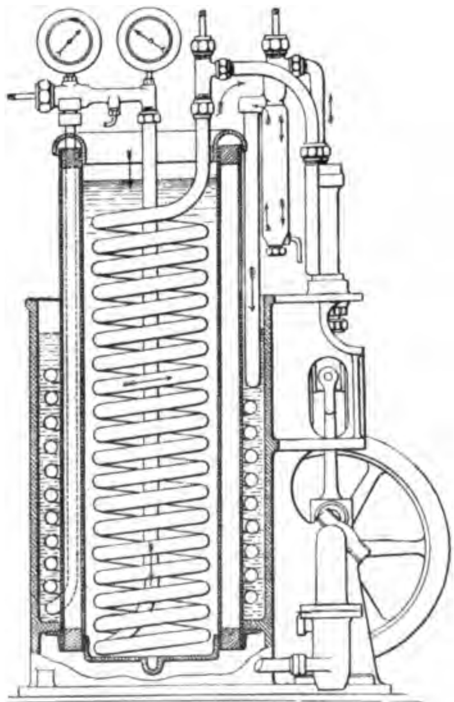


FIG. 12. —SECTION OF SMALL CARBONIC ACID MACHINE.

copper pipes for condensers, the many advantages of copper over iron when subjected to the action of sea-water will always be a heavy handicap for ammonia machines at sea.

In a paper read before the Ipswich (England) meeting of the British association on "Carbonic Anhydride Machines," by Mr. Hesketh, one of the directors of Messrs. J. & E. Hall, Ltd., of Dartford, a firm that has introduced these machines all over the world, it is clearly shown that with a machine

producing 9,360 pounds of ice per twenty-four hours from water at the various temperatures tabulated below, the inlet water for the condenser being the same, the indicated horse power varied as follows:

Temperature of water.....	52°	75°	85°	90°	100°
I. H. P. of engine.........	15.62	20.03	27.2	28.2	42.10

From a series of experiments made by Messrs. L. A. Riedinger & Co., of Ansburg, the following results were deduced:

Temperature condensing water ...	55° to 69°	95° to 97.7°
Ice production per hour	485 pounds.	257 pounds.

Other tests of cooling brine show that with its temperature reduced to 50° and to zero, and with condensing water at 60° and 100°, the efficiency was reduced in one case 40 per cent and in the other 60 per cent by the use of the warmer condensing water.

As a contrast to these results the relative efficiency of a cubic foot of ammonia gas under different temperatures between 65° and 105° is shown by the following table, the figures representing the refrigerating effect in thermal units, as given by Professor Siebel in the "Compend of Mechanical Refrigeration":

Gauge suction pressure in pounds,.. / Corresponding tem. in refrigerator...		4 / —20°	9 / —10°	16 / 0°	24 / +10°	33 / +20°	45 / +30°
Temp. Fahr.	Gauge condenser pressure in pounds.	Refrigerating effect of a cubic foot of ammonia gas in British thermal units.					
65°	103	33.74	42.28	54.88	68.66	85.15	106.21
75°	127	33.04	41.41	53.76	67.27	83.44	104.09
85°	153	32.34	40.54	52.64	65.88	81.73	101.97
95°	184	31.64	39.67	51.52	64.49	80.02	92.85
105°	218	30.94	38.80	50.40	63.10	78.31	97.73

Showing that with back pressure from four to forty-five pounds the increase of condenser temperature from 65° to 105° only reduces the efficiency of a cubic foot of gas about 9 per cent.

CHAPTER VIII.

THE ABSORPTION SYSTEM.

Although compression machines now largely out-number those working on the absorption principle, it must be remembered that the latter led the way and for a long time carried all before them. Introduced in 1858 by Ferdinand Carré, of France, and in 1861 into Australia by Mr. E. D. Nicolle, this system was largely developed by the skill of that gentleman and the munificence of the late Mr. T. S. Mort. By the erection of ice works at Darlinghurst in 1863-64 the ammonia system supplanted the ether machines of Harrison in New South Wales at about the same time as Reece and others were working out the same problems in Europe. At the Darlinghurst works food was kept in cold storage for fifteen months, from the end of 1865 to 1867, when the plan shown by Fig. 13 was prepared by Mr. Nicolle, and seems to be the first authenticated proposal ever made for the purpose of refrigerating on shipboard.

Mr. Nicolle is now seventy-five years of age, and has for many years retired from active business; he still, however, has a great disbelief in compressors, and has been for three years past working at his beautiful country home, on Lake Illawarra, developing a new process which he hopes soon to make public, with the result—to use his own words to the author—"of leading this interesting art into its proper channel again."

Under the absorption system an aqueous solution of ammonia is the medium used, instead of pure anhydrous ammonia. Taking a solution of twenty-five parts of ammonia in seventy-five parts of water, in a boiler or still, the application of heat will cause both gas and aqueous vapor (steam) to be given off in the proportion of, say, 90 per cent of ammonia

gas to 10 per cent of steam or vapor. This combined vapor is passed into a condenser under the pressure maintained in the boiler or still, and such pressure is mainly dependent upon the temperature and volume of the condensing water.

As an effect of this pressure and the transfer of heat to the condensing water the ammonia is liquefied. This liquid ammonia is then allowed to expand in the coils of the refrig-

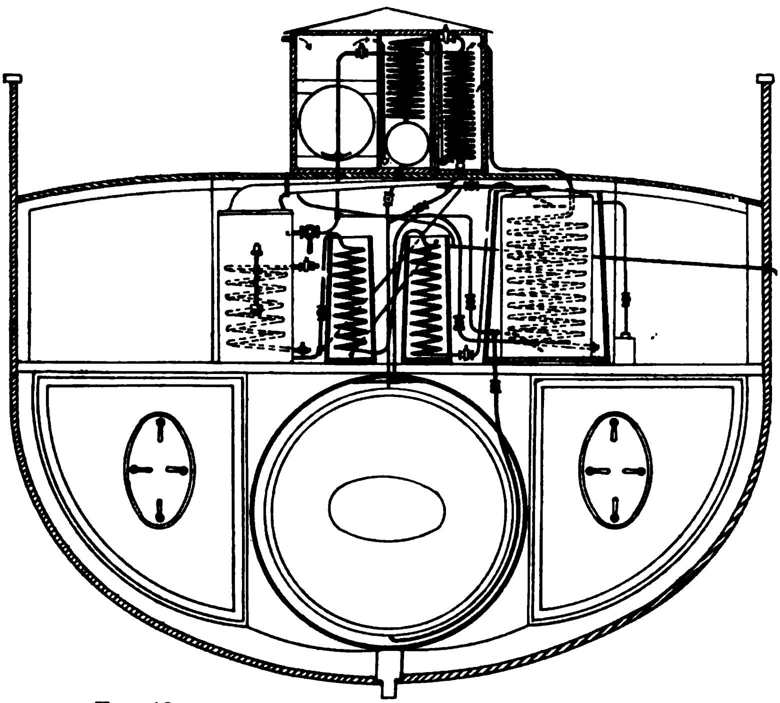


FIG. 13.—SECTION OF SHIP FITTED FOR COLD STORAGE.
Planned by Mr. Nicolle, 1867, for the shipment of meat from Australia to England.

erator, where it either freezes or cools the substance it is employed to refrigerate. The gas being driven out of the boiler or still by the pressure generated, the solution left—called the weak liquor—is then drawn out and cooled in another condenser, after which the ammonia from the refrigerator and the weak mother liquor are allowed to re-unite and form strong liquor in a vessel termed the

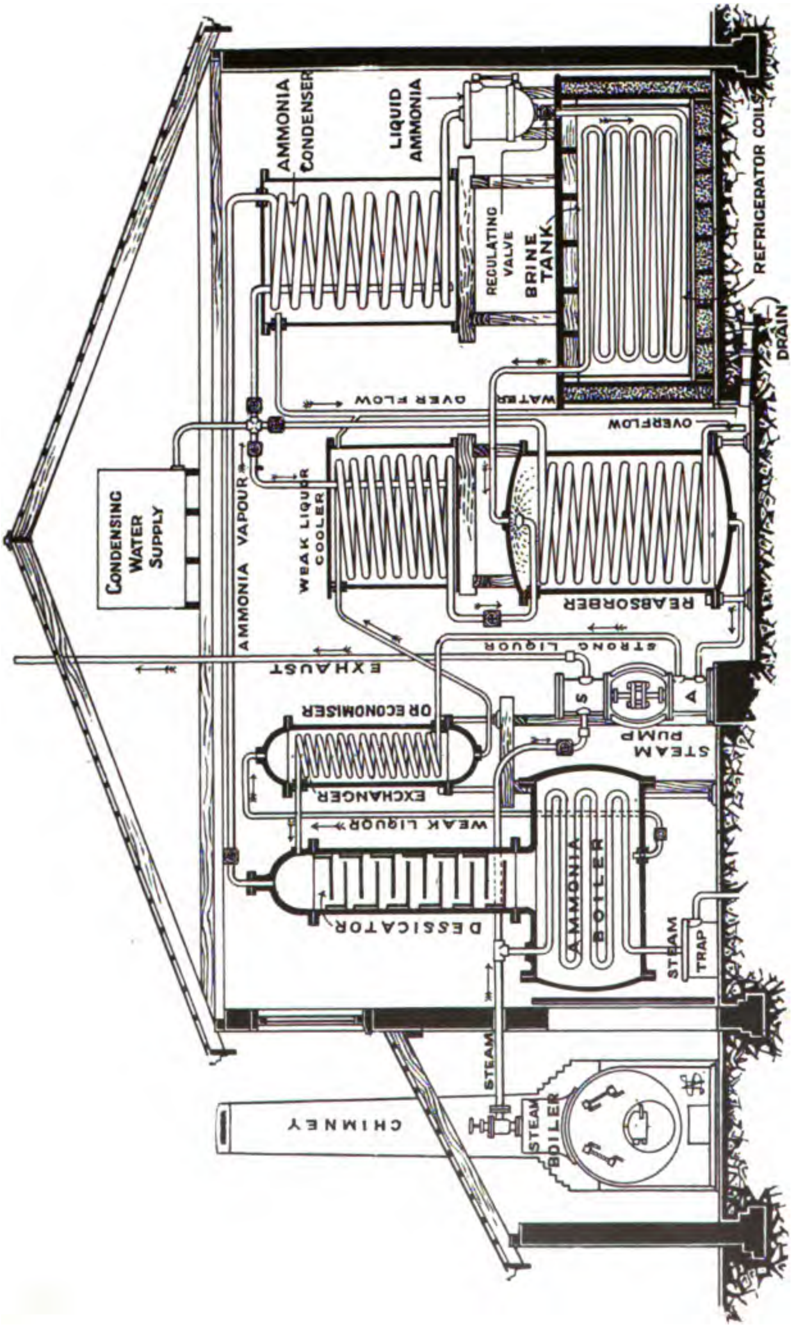


FIG. 14.—DIAGRAM REPRESENTING CYCLE OF OPERATIONS IN AN AMMONIA ABSORPTION SYSTEM.

absorber, from which the system takes its name. After this the strong liquor can be returned to the boiler to go through the same cycle of operations, which may be repeated over and over again.

Fig. 14 is an illustration—not drawn to scale—which will enable the whole process to be comprehended; from this the great importance of the desiccator and exchanger will be

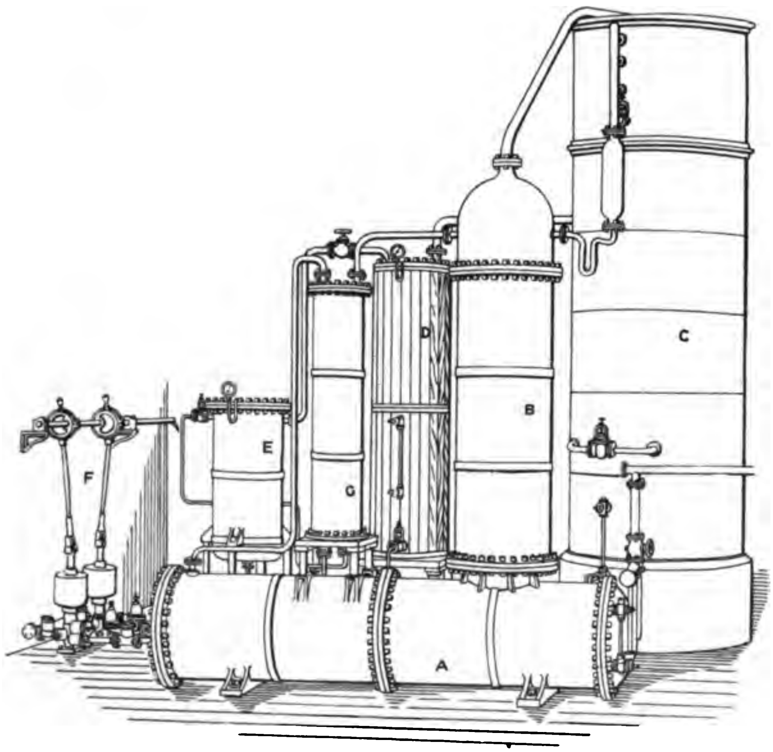


FIG. 15.—AMMONIA ABSORPTION PLANT--ENGLISH PATTERN.

understood—the former, by its separation of watery vapor or steam from the hot gas, saves fuel and condensing water directly; and the latter, by transferring the heat from the weak liquor (which has to be cooled before it again absorbs the gas) to the strong liquor (which is coming back to the boiler to have the ammonia driven off again), saves fuel indirectly as well as condensing water.

The absorption system involves a comparatively simple process, because the apparatus required consists mainly of the several vessels, pipe coils and valves, and there is no motive power or moving machinery required except the pump to return the liquor from the absorber to the boiler. Even the pump can be dispensed with, as under several ingenious arrangements a vessel like the "Montejus," used in sugar mills, is employed, by which the strong liquor is lifted by the pressure of the gas to an elevated receiver and descends to the still by gravity.

Fig. 15 is a perspective sketch of an English absorption machine which has been largely used in breweries.

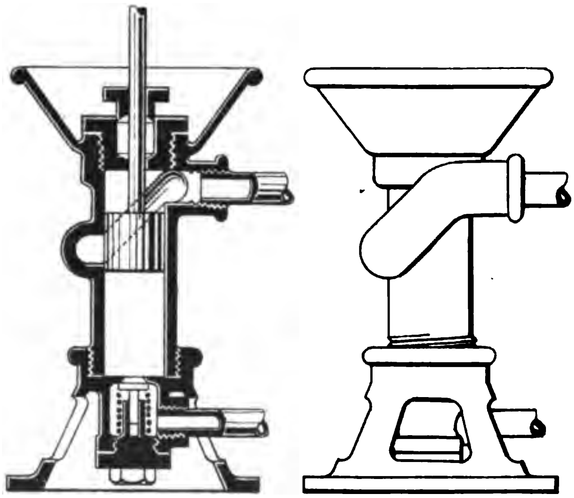


FIG. 16.—PUMP TO FILL BY GRAVITY FOR AQUA AMMONIA.

Under one of the patents taken out by Messrs. Mort and Nicolle in New South Wales there were two re-absorbers used which worked under pressure and vacuum respectively, and in order to overcome the difficulty of withdrawing the liquor from the vacuum chamber the pump shown by Fig. 16 was specially designed by the author.

The class of machinery used being relatively cheap as compared with compressors, and the process being a simple one, absorption machines are still made and used under certain conditions. Since its first adoption many elaborations

have been made to the original elements in order to secure fractional distillation and desiccation of the gas, and also by means of larger exchangers to utilize more of the waste heat; but perhaps the greater amount of condensing water required rather than the greater quantity of fuel practically, if not theoretically, wasted in the absorption machine is the reason that the compression system has taken the lead in popular favor.

It is found that water at atmospheric pressure and 60° F. will absorb about 700 times its volume of ammoniacal gas, and that watery vapor will often distill over with the gas, which largely discounts the efficiency of the machine, because this vapor not only requires fuel to raise it, but a supply of cold water to condense it, and although the increased amount of fuel required might not condemn the use of the absorption system where fuel is cheap, yet in most parts of Australia, having to supply double the quantity of condensing water would be a serious drawback, and has led to an increased demand for compression machines.

Many changes and improvements have been made in the construction and mode of operation of absorption machines in America recently, some of the latest types of which are illustrated and described in Chapter XX of this book.

CHAPTER IX.

THE COMPRESSION SYSTEM REVERTED TO.

As soon as the defects of the absorption system were understood inventors reverted to the work of Perkins, Harrison and Twining, but it was found to be a very different matter to compress a subtle gas like ammonia up to twelve or more atmospheres than it had been to deal with ether vapor at a comparatively low tension; and the results now attained have only been reached by a long series of experiments which had for their object the improvement of the compressor. Toward this work English, American, Continental and Australian inventors have all contributed. When we come to compare the machines of different makers, we shall find that great diversity of opinion exists with regard to details, and that many of them keep certain special points in view to the neglect of others which are not, in their opinion, of so much importance; hence we have a large choice of ammonia compressors in the market, some of most admirable design and workmanship, and nearly every one of their respective agents claims for his machine that it is the best in the world. As it is hardly possible that they can all be the best absolutely, seeing how widely they differ from one another, it will be instructive to take a few of the leading types and, comparing one with the other, examine into their construction, method of operation and relative efficiency.

It must be admitted that theory and natural laws have no favorites, and that the conditions which result from compression and expansion are the same for every one; but theory alone is of little avail in the work of the mechanical engineer, and some of the biggest failures in practice have resulted from hugging one main central theory so closely that all the little attendant theories were forgotten. The

practical experience which ensures success generally carries with it a knowledge of many little theories which the ordinary theoretical man or mathematical expert has no opportunity of making acquaintance with, and it has been mechanical

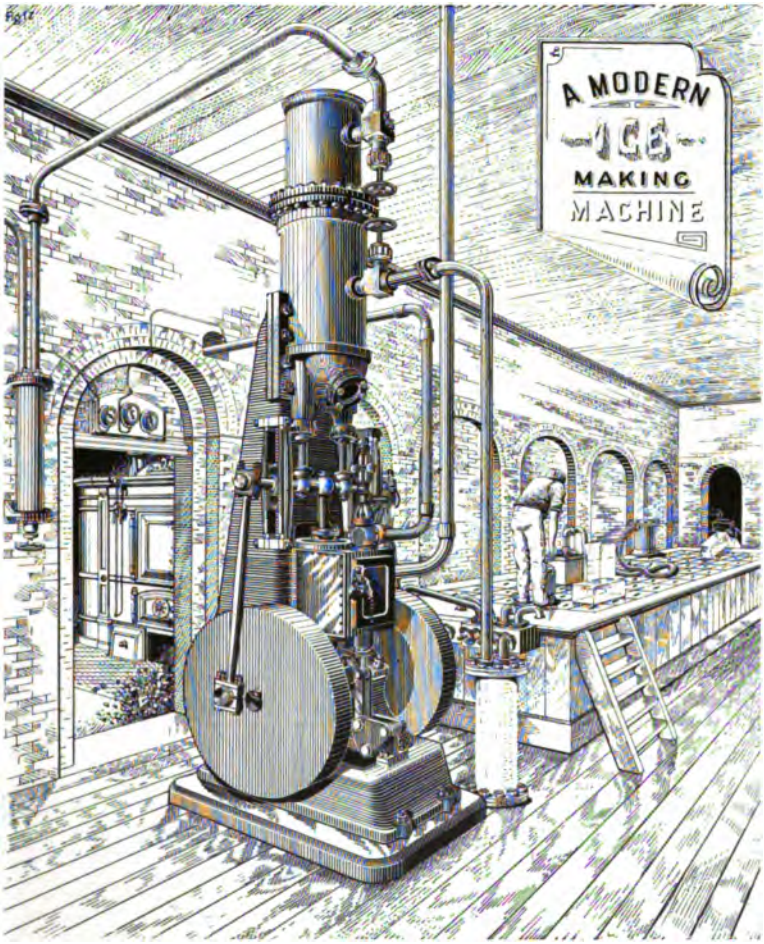


FIG. 17.—PERSPECTIVE VIEW OF AUSTRALIAN ICE MAKING PLANT.

engineers rather than mathematicians who have brought the ammonia compressor to its present improved state.

Fig. 17 shows an ice making plant recently built in Australia, where an endeavor has been made to produce a com-

pressor that should embody as many good features as possible by profiting from the experience gained with many of the well tried designs already in use there, which have been made in America, England and Germany respectively.

ALL COMPRESSION SYSTEMS EMBODY THE SAME GENERAL PRINCIPLES.

Reference has been made to the many admirable books which are published as trade catalogues by makers of refrigerating machinery, and there are undoubtedly among those which refer to compression plants for ammonia many which are noticeable for the excellence of their illustrations and the amount of information which they make public. Some of these works, however, speak of *our system* and "the *principles* on which *our machines operate*," in a way that might be taken to imply that such systems and such principles were special and uncommon, whereas they are generally exactly the same as those adopted by other manufacturers in the same line. The special characteristics of the leading makers of machinery are now generally confined to improvements in details.

The use of anhydrous ammonia and of apparatus for the liquefaction of its gas is common property, and a compression plant of the present day embodies exactly the same four fundamental sections which Jacob Perkins showed in his 1834 patent (see Fig. 1); that is: (1) The refrigerator, where the medium is vaporized by the heat given up; (2) the pump to withdraw the vapor or gas from the refrigerator and compress it into (3) the condenser, where it is cooled and liquefied; and, lastly (4) the regulating cock or valve, by which the admission of the liquefied medium into the refrigerator again is regulated. These four leading features are amplified in modern plants by appliances for forcing oil into the compressor cylinder, or to the stuffing-box of the piston rod; by special devices for separating oil and foreign matters from the medium; by the use of vessels for storing the liquid refrigerant, and so on.

While there is great diversity to be found in the practical construction of condensers, refrigerators and other appurtenances that will be referred to in their place, all of these appliances put together do not seem to have afforded so much

scope for originality of design, as well as diversity of arrangement and construction, as the compressor itself.

RELATION OF THE SEVERAL PARTS OF A REFRIGERATING MACHINE TO ONE ANOTHER.

The relation which the refrigerator, the compressor and the condenser of a refrigerating plant occupy with regard to one another is much the same as that which exists between a steam boiler, an expansion steam engine and a surface condenser—each section of the apparatus in either series begins and completes its work upon the medium employed without its efficiency being dependent upon either of the others. As the efficiency of any steam engine is entirely independent of the kind of boiler which supplies it with steam, and the efficiency of the boiler is not measured by the engine, so in a refrigerating plant no particular form or arrangement of condenser or refrigerator is necessarily coupled with any special design of compressor. Individual makers of machinery, however—no doubt for sufficient reasons—often appear to prefer and certainly do adopt certain special combinations of details as their own, but this does not affect the argument that such is not indispensable for successful work.

Given ample surface for the conduction of heat, plenty of section for the gas to pass without friction, a free get-away for the liquefied ammonia, or other medium, as cold as possible, and absolutely tight joints, it is more a matter of cost and convenience rather than of efficiency whether the tubes of a condenser are of small or large diameter, straight or coiled, horizontal or vertical, or even whether the condenser and refrigerator are made of tubes at all. The first ammonia refrigerators, made in Sydney about the year 1860, were flat boxes constructed of boiler plate, closely stayed like the walls of a locomotive fire-box, and they were effective; but a coil of tubes electrically welded, such as is now procurable, would not only be more convenient, but, for a given surface, would cost only a small fraction of the amount that the stayed boxes did.

CHAPTER X.

IN THE LIQUEFACTION OF A GAS THE WORK OF THE COMPRESSOR OR PUMP IS SUPPLEMENTED BY THE ACTION OF A CONDENSER OR COOLER.

THREE KINDS OF CONDENSERS ARE USED.

Refrigerating condensers may be divided under three separate heads. *First*, The "submerged," having coils generally arranged spirally and immersed in a tank of water. *Second*, The "atmospheric," having the coils more commonly made of straight lengths of tube with return bends, all exposed to the air, with a trickling of water constantly flowing over them; and *Third*, The "evaporative," similar to the atmospheric in general arrangement, but with the addition of devices to promote the rapid evaporation of a smaller water supply from the external surfaces.

Submerged Condenser.—When there is an unlimited supply of water the submerged condenser has certain advantages, one of which is that the cold water can enter its tank near the exit of the condensed gas at the bottom, rising as it becomes warmer to where it overflows, and thus, by having the gas delivered into the top ends of the tubes, its downward flow is in the opposite direction to that of the water, and the exit of the liquefied gas is in the coldest part of the condenser—at the bottom. Besides this, in most waters the pipes keep clean longer if fully immersed. To make a submerged condenser thoroughly efficient the water should be kept mechanically agitated all the time it is at work, otherwise a film of warm water forms around the pipes and prevents the full transference of heat to the gradually rising body of water which overflows at the top of the condenser tank. Fig. 18 shows a condenser of this description of Eng-

lish make (it is the left hand vessel, which is in section), the four spiral coils for the ammonia and the helical-bladed agitator with its driving wheel being clearly indicated. The right hand vessel is similar in general construction, but is a refrigerator or cooler. In condensers or refrigerators of this description the agitation should only be sufficient to keep the water moving past the surface of the coils, and should not break up the zones of temperature by setting up vertical

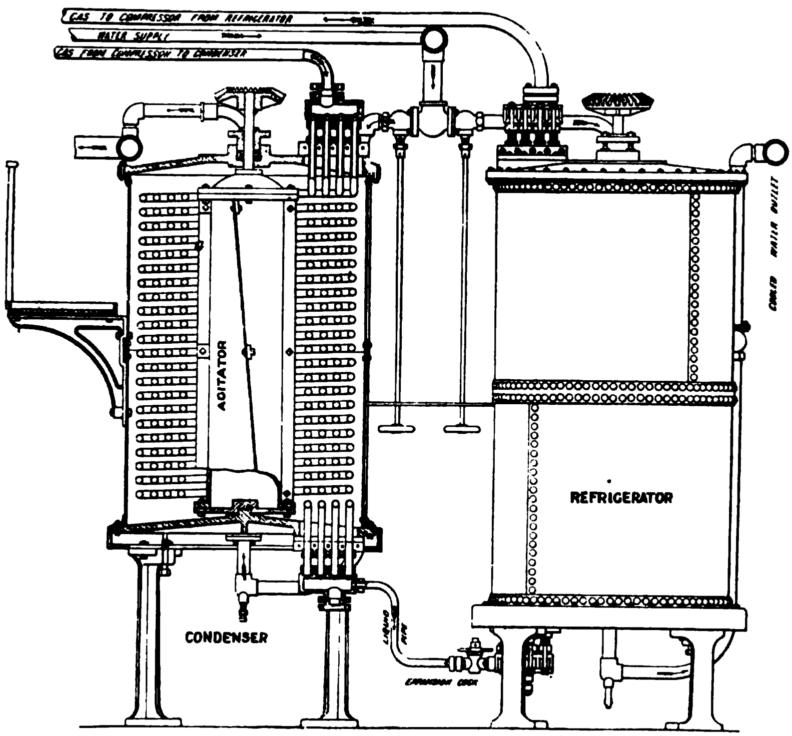


FIG. 18.—SUBMERGED CONDENSER—ENGLISH PATTERN.

currents, because in such a case, by making the temperature more uniform throughout, the liquid ammonia would not be cooled so much and the water would go away cooler.

Atmospheric Condenser.—An ordinary form of atmospheric condenser is seen in Fig. 19, which shows a stack of fifty-six tubes in four lines, with cast return bends and heads, and having four water distributors at the head.

With a condenser of this description the evaporation of the water flowing over it may be so great in very dry climates and under certain conditions as to enable it to be used over and over again less the loss due to evaporation. On the dry plains of Riverina* during a gentle breeze, water at 90° flowing on to an ammonia condenser has been known to leave the bottom coils several degrees lower in temperature, the cooling effect of atmospheric evaporation more than compensating for the heat taken up from the ammonia.

As the water must have a downward flow over the pipes in this class of condenser, the gas must enter at the bottom and ascend as it is being condensed if it is to travel in the

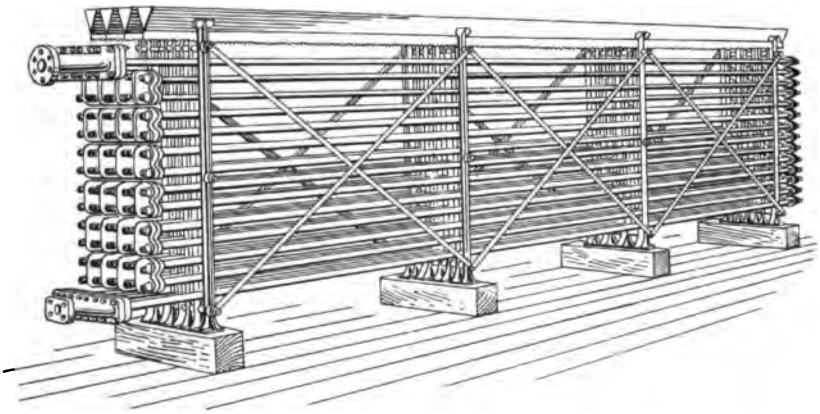


FIG. 19.—ATMOSPHERIC CONDENSER—AMERICAN PATTERN.

opposite direction to the water. This arrangement, of course complicates the collection of the liquefied ammonia, and in the condensers of some eminent makers who adopt this plan they provide for the interception of the condensed medium by connecting small pipes at every alternate bend of the condenser which carry off the ammonia directly it is liquefied to the liquid storage tank, as shown in the general arrangement of a De La Vergne plant, Fig. 20.

In most atmospheric condensers of moderate size, however, the gas enters at the top, and with the condensed liquid has a continuous downward flow. In order to get the benefit

* The country between the Murray and Murrumbidgee rivers in Australia.

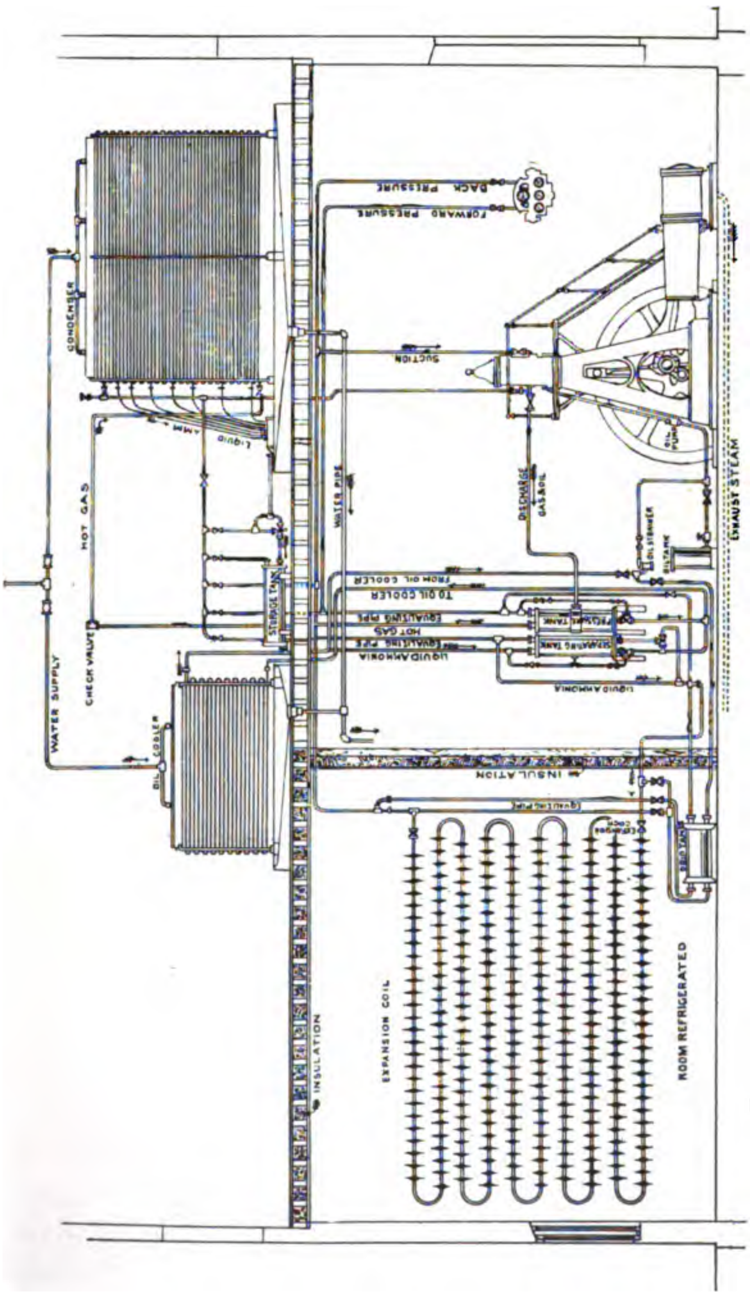


FIG. 20.—DIAGRAM ILLUSTRATING CYCLE OF OPERATIONS IN A DE LA VERGNE PLANT.

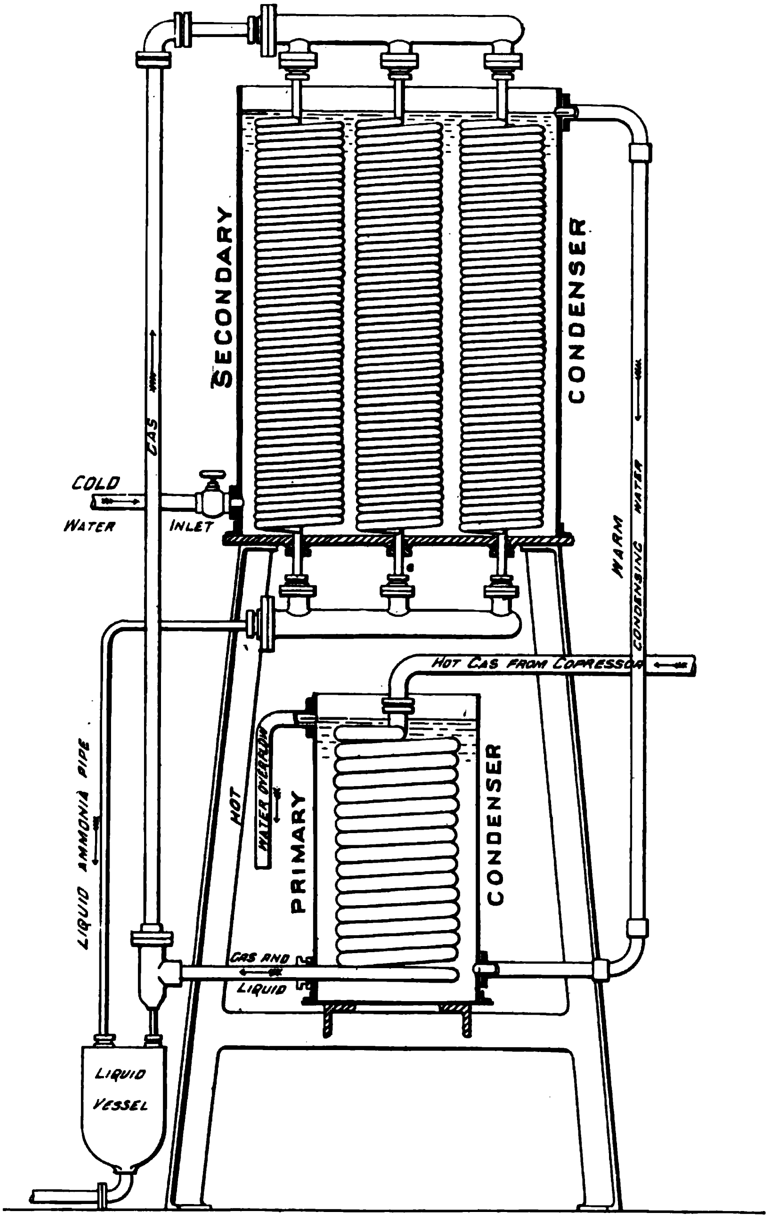


FIG. 21.—COMPOUND SUBMERGED CONDENSER—CORRECT PRINCIPLE.

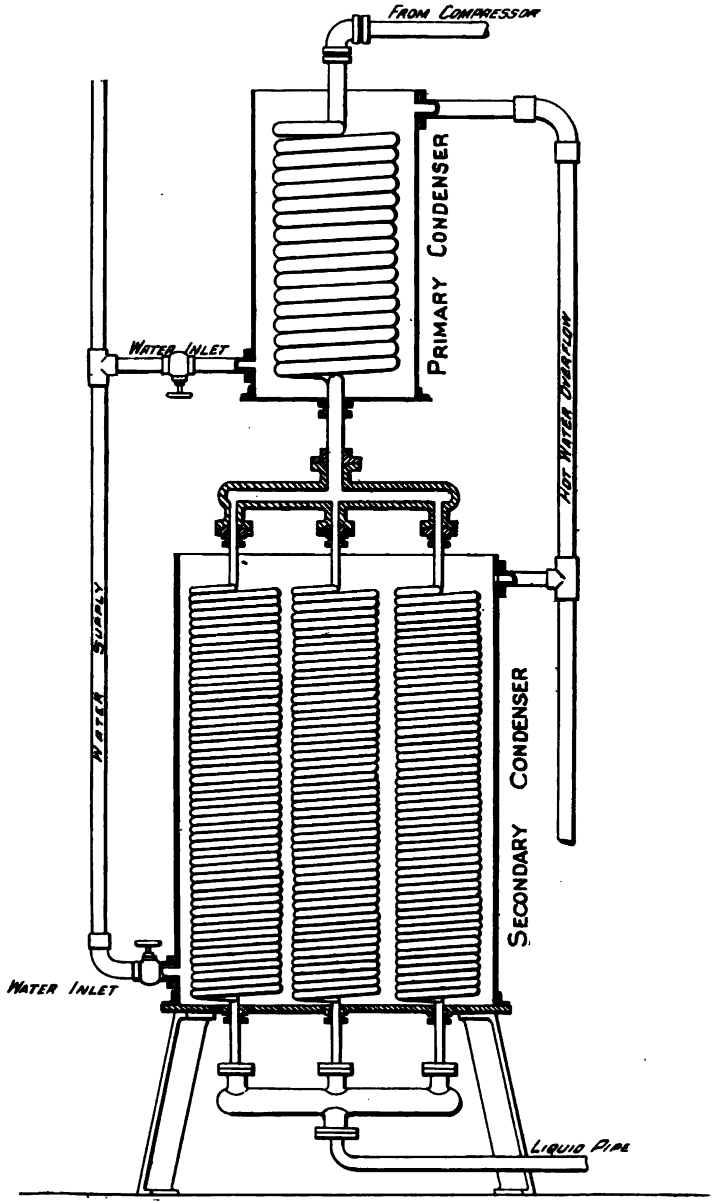


FIG. 22.—COMPOUND SUBMERGED CONDENSER—WRONG PRINCIPLE.

of cold water to the later stages of condensation, and thus reduce the liquid medium as low as possible, while both the external and internal fluids have a downward flow with regard to the coils, several devices have been adopted. In one of these a primary condenser is submerged in a tank which is fed by the overflow from an atmospheric condenser above, and the medium, after being cooled in the lower coils, passes up again to the top of the colder condenser overhead. The advantages of such an arrangement when the scale of the plant warrants it are obvious. In other cases builders adopt two or more stages of submerged condensers, sometimes as in Fig. 21, and at other times as in Fig. 22; but it is not quite clear how any gain can result from the increased complication in the latter case, where each tank has a separate water supply in parallel, and the proper arrangement to save water is as Fig. 21, which shows the same condensers with a water supply in series. Fig. 23 shows a two-story atmospheric condenser designed by the author for hot climate and scarcity of water, in which the gas flows down through the lower coils first and then passes from the bottom right up to the top of the upper coils, the liquid being drawn off separately at the bottom of each coil. In such an arrangement the upper coils may be of smaller tubes than the lower ones.

Evaporative Condensers.—If the coils of an atmospheric condenser are covered with a light fabric which is kept wet, while an artificial current of air, propelled by a fan, is passed through them, so that a powerful evaporation is set up, it is possible for the water to be as cool at the bottom as at the top, just as in the instance occurring naturally in a specially dry climate before referred to. In such cases the gaseous and liquid medium may flow downward and still have its final cooling effected by the minimum temperature of the condensing water. The economy or otherwise of such an arrangement depends entirely upon the cost per gallon of the water and its initial temperature, as compared with the cost of the power required to drive the fan.

THE RE-USE OF CONDENSING WATER.

Although submerged condensers require a large supply of water they are often used where water is costly,

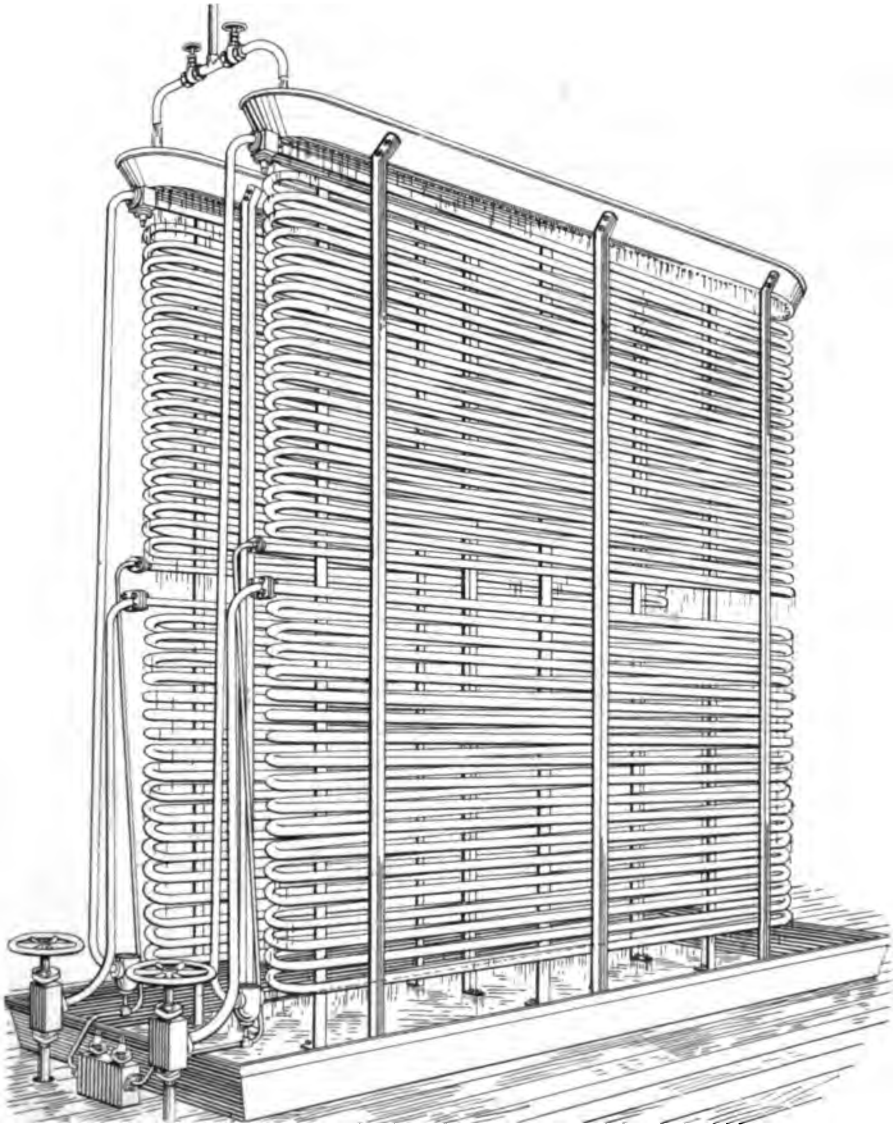


FIG. 23.—TWO-STORY WATER SAVING ATMOSPHERIC CONDENSER—AUSTRALIAN PATTERN.

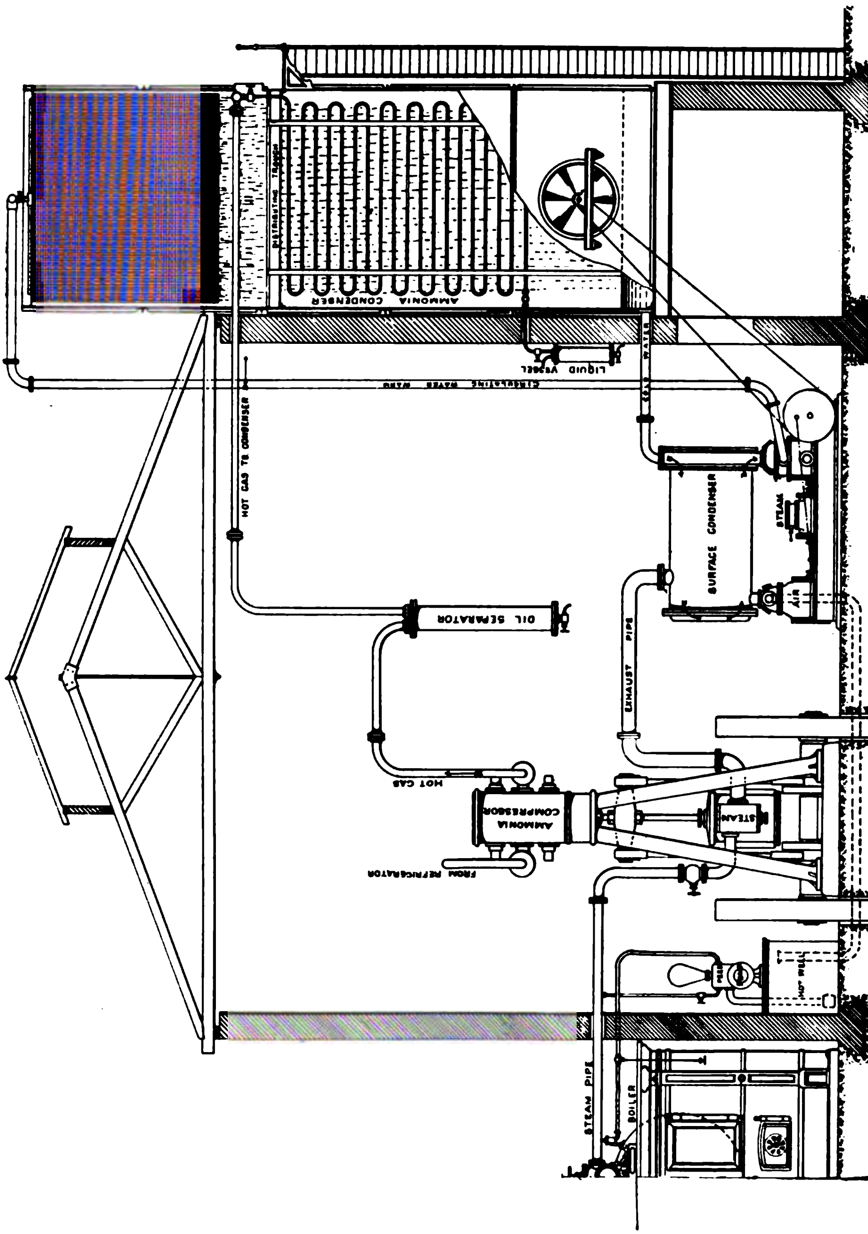


FIG. 24.—REFRIGERATING PLANT WITH AMMONIA CONDENSER, SURFACE CONDENSER TO STEAM ENGINE, AND COOLING TOWER PROVIDING FOR RE-USE OF CONDENSING WATER.

but under an arrangement by which the water is used over and over again. There are many arrangements differing in detail by which this may be effected, but they all turn upon the transfer of the heat taken up by the water, partly to the atmosphere and partly through the evaporation of a portion of the water itself. The simplest arrangement, and the one in most common use, is probably a louvred tower through which the air circulates while the water descends in a rain or spray, the shower in some cases being broken up by baffles consisting of layers of foliage or by screens of special mechanical construction.

In other cases the water is played upward from innumerable fine jets over a water-tight floor, and the diffusion into the finest spray brings every particle into contact with the air. In special cases evaporation is accelerated and cooling is effected by an upward current or blast of air from a fan or air propeller through a tower with closed sides, which is stacked in some cases with porous pottery or metal pipes and in other cases with sheets of woven wire cloth. The water is sprayed at the top of the tower by a Barker's mill arrangement or by perforated pipes, and is divided and subdivided at every separate layer by the obstructions placed for the purpose. These processes of cooling are used in other industries than those connected with artificial refrigeration, notably for condensing steam engines which require a continuous supply of cool water. A very full description of them would, therefore, be rather beyond the scope of this work.

Fig. 24 is an arrangement of an evaporative condenser in a cooling tower erected at such an elevation that the water flows direct to the surface condenser of the steam engine, and the whole circulation is maintained by one pump coupled to the air pump, the engine for which also drives the fan.

CHAPTER XI.

THE REFRIGERATOR.

The refrigerator, which corresponds to the boiler in a steam engine system, generally consists of a series of tubes, through the metal of which the heat abstracted from the substance being cooled is conducted to the medium which flows through them, and this heat is transferred to the medium under two distinctive systems of practical refrigeration.

Under the first or brine system, as it is termed, there are coils of tubes arranged in a way similar to those of the condenser (see left hand vessel of Fig. 18) which are immersed in a tank of non-congealable liquid, generally a solution of ordinary salt (chloride of sodium) or chloride of calcium; from this liquid heat is taken up by the refrigerating medium. This brine derives its heat either from vessels which are immersed in it, as when ice is to be made, or from the atmos phere which surrounds it, as when chambers are to be refrig-erated. In the latter case tubes or troughs are placed in the air of the chamber through which the cold brine flows, and this cold brine abstracts the heat from the room.

Fig. 25 represents an ice making tank as filled with brine in which the ice molds are inserted. The centrifugal pump at the right hand side draws the brine from under the false bottom and delivers it over the top of the end diaphragm, and so creates a perfect circulation, which can be controlled by regulating the openings in the false floor. The expansion coil is shown as in one length, the only joints being the flange connections to the manifold expansion and return valves. Instead of having a centrifugal pump for circulation within the tank, it is evident that a force pump could be employed to circulate the brine through a series of pipe coils on the walls or

ceiling of a chamber, in order to withdraw the heat from the same and its contents.

Under the second or direct expansion system the gas in the refrigerator coils takes up heat by direct conduction from the air of the rooms to be cooled. This transference of heat may take place in the cold chamber itself, over the walls or ceilings of which the expansion pipes may be laid. Fig. 20 shows how the De La Vergne Company increase the surfaces of these pipes by stringing on them a series of discs to act on the same principle as the "gills" of heating apparatus. Air may also be cooled by direct contact with the surfaces of the refrigerator in a *separate* chamber, and then be made to flow into the rooms to be cooled by a natural or forced current. This was the subject of a long since expired patent by the author.

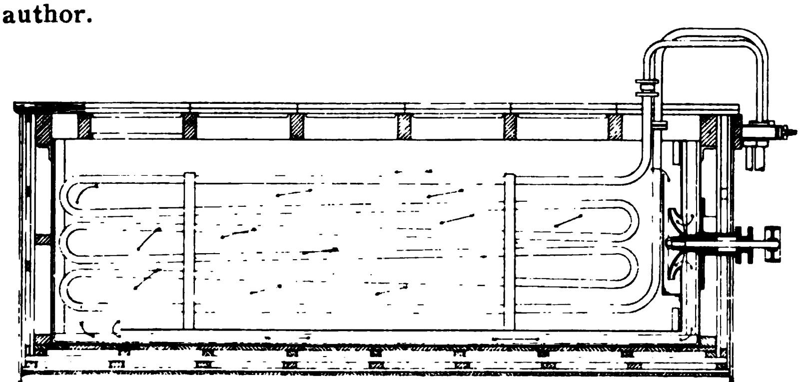


FIG. 25.—BRINE REFRIGERATING TANK AND CENTRIFUGAL AGITATOR.

In another system of cooling chambers, which is extensively adopted by the Linde company, the refrigerator cools the brine, and the brine cools a series of iron plates, alternately immersed in and withdrawn from it, which are arranged as revolving discs. These metallic surfaces cool the air which circulates between them, and transfers to the brine from the air the heat which it has abstracted from the goods to be refrigerated. In this latter case there is a fourfold transference of heat and consequent loss of power, besides a great drying action, but there are compensating advantages arising from the ease with which the circulation of the air can be controlled and directed by channels

wherever required. With direct expansion there is only a double transfer of heat, that is, from the goods to the air and from the air to the refrigerator; and on these grounds it is the more economical arrangement, especially as regards expenditure of power for a given amount of heat abstracted. The brine system has its own advantages in special cases, one of which is the great facility which it affords for storing negative energy by having large tanks of cold brine in reserve. Such reservoirs, by their capacity for taking up heat, can for more or less time be used in case of stoppage of the machine. Another advantage claimed for the brine system is the absence of danger by the escape of ammonia into the cold chambers.

With the very perfect system of jointing pipes now adopted by the best refrigerating engineers, however, there is probably more of sentiment than reality underlying the fear of danger from leakage of ammonia which is felt by some persons with direct expansion. The brine system affords greater facilities for subdividing the cooling power of a large machine among a great number of separate operations by reducing the care and attention required at the expansion valves, but an expert would hardly decide whether to use brine or ammonia circulation, or both combined, in any particular industry involving the use of artificial cold until he had the whole of the requirements before him.

CHAPTER XII.

THE SURFACE REQUIRED FOR EXCHANGE OF TEMPERATURES IN CONDENSERS AND REFRIGERATORS.

There may be scope for a great deal of personal predilection in connection with the various patterns of condensers and refrigerators thus far referred to, and which manufacturers avail themselves of to the fullest extent, often influenced, perhaps, by a reputation attached to their "system," and also by their available tools and appliances. Further than this any firm having once settled on a particular pattern or method of construction would be loath to change it, if the advantages of doing so were at all open to question.

When, however, it comes to the amount of surface to be provided for the conduction of heat that has to be transferred, and the sectional pipe area for the passage of the gas, we still find that personal fancy and "rules-of-thumb" largely prevail, although the *pros* and *cons* admit of more exact calculation, and the effect of any variation in the proportions of such parts is more easily seen and understood.

Now the function of all condensers and refrigerators is to transmit heat from one substance to another, generally from a gas or vapor to a liquid, or *vice versa*, and through the walls of the apparatus. The amount of heat so transmitted is principally dependent upon two conditions, which are: *First*, The difference in temperature of the two substances; and, *Second*, The superficial area of the surfaces of transmission. To a large extent also the result is dependent upon the velocity at which the gases, vapors or liquids move over the condensing surfaces, in a lesser degree on the relative conductivity of the substances themselves, and to some extent on the character of the metal walls of the appa-

ratus. If it were not that in practice these metal walls are relatively very thin the conducting power of the metal would be of much greater importance than is actually the case.

It may be taken as an axiom that the amount of surface necessary in any condenser or refrigerator is *directly* as the number of units of heat to be transmitted, and *inversely* as the difference of temperature which is permissible between the two substances. The importance of having sufficient surface thus becomes apparent if it is desired to cool as low as possible, and to utilize the maximum amount of the machine's work.

The walls of the condensers and refrigerators are almost invariably now of metal tube, copper being "taboo" for use with ammonia. The relative conducting powers of the principal metals, taking gold as a standard, are as follows:

Metal.	Relative conducting power.	Metal.	Relative conducting power.
Gold	1,000	Cast iron	562
Platinum	981	Wrought iron	374
Silver	973	Zinc	363
Copper	892	Tin	304
Brass	749	Lead	180

It will be noted from the above that wrought iron, the material usually employed for ammonia, is at a considerable disadvantage when compared with copper, which can be used with carbonic anhydride and sulphur dioxide machines. This difference of 374 to 892 is, relatively, very great, and would require serious consideration if large masses of metal were concerned. Actually it is of small importance in tubular condensers, owing to the metal being so thin that it is practically at the same temperature on both sides of the tube.

With regard to the conducting power of the gases themselves, there do not seem at present to be any records available that have been obtained by means of actual trials with working machinery, and carried out with exact instruments in the hands of careful observers.

Laboratory experiments have been carried out by Professor Magnus upon the four following gases: Atmospheric air, hydrogen, carbonic acid and ammonia. A large tube was inserted in a glass flask containing water at the boiling point, a delicate thermometer was fitted in the center of this large

tube, and smaller tubes enabled the larger one to be filled with the several gases. The time was noted which was required for heat to be transmitted through the several media, with the following results:

Name of Gas.	Rise of Temperature.	
	From 20° to 80°	From 20° to 90°
Atmospheric air	3.5 minutes	5.25 minutes
Hydrogen	1.0 "	1.4 "
Carbonic acid	2.25 "	6.3 "
Ammonia	3.5 "	5.5 "

It will be seen from the above that hydrogen shows an extraordinary power of conduction, and that carbonic acid is sluggish, while the conditions appertaining to ammonia seem to correspond so closely to those of air, that the tables which have been obtained from experiments made on heating air by hot water pipes may possibly be sufficiently accurate for all practical purposes, if applied to the parallel operation of heating ammonia gas in the coils of a refrigerator.

According to Box the loss of heat from the contact of air with cylinders two inches in diameter is .728 units per square foot for one degree of difference, the efficiency falling with larger pipe and rising as the difference of temperature increases. When the difference of temperature reaches 150°, more than two units, instead of .72 of a unit, is transmitted for every square foot of surface and degree of difference. On page 128 (third edition) of the "Compend of Mechanical Refrigeration," this factor (M) is given as .5 unit without any reason being assigned, and if initial cost is of less importance than permanent efficiency, it is certainly taking the safe side to make it so low in figuring out for either a condenser or refrigerator.

When hot, dry gas is cooled down and becomes a saturated vapor one would suppose that the data obtained from the surface condenser of a steam engine would be most applicable to the case of proportioning the condensing surface for refrigerating machines. From some experiments made by Mr. Nichols, recorded in D. K. Clark's large manual, the following results appear and show the heat transmitted both with horizontal and vertical tubes, and also with differ-

ent velocities of condensing water flowing over their surfaces:

	Vertical Tubes.			Horizontal Tubes.		
Velocity of condensing water in feet per minute........	81	279	390	78	307	415
Heat transmitted per hour per sq. foot for each degree difference in T. U. ..	295	383	401	422	530	600

The radiating or absorbing power of iron, according to Peclet, equals .56 of a B. T. U. per square foot for each degree Fahrenheit difference in temperature, but it is evident this general statement is of no value for practical application.

The following table shows how the conducting power of cylinders falls off as they increase in diameter from two inches to eight inches, the units being the number transferred per square foot for each degree difference in temperature:

Diameter in Inches.	Heat in Units.	Diameter in Inches.	Heat in Units.
2	.7280	6	.5230
3	.6256	7	.5087
4	.5747	8	.4978
5	.5440		

As this table does not include cylinders less than one inch diameter, the ratios actually given have been utilized in constructing a curve, from which it appears that a cylinder one inch diameter, or say three-fourths inch iron pipe, would probably transmit .84 or .85 of a unit per square foot for 1° difference.

The data given all go to show that the preference for small pipe is established on bed-rock truths, and they further suggest that possibly the liquid ammonia is often withdrawn from the condenser at the temperature of liquefaction through being run off at once to the liquid vessel as soon as it is condensed, when it might have been cooled a few degrees lower with advantage if left in the condenser longer.

Refrigerating authorities have deprecated the use of the bottom coils of the condenser as the permanent and only liquid vessel, and with good reason, but it is possible that if a very short extra coil of small pipe between the gas condenser and the liquid bottle was so arranged as to be always kept full of the running liquid, either by means of a siphon or some

other device, it would enable the liquid to be brought down to within 1° or 2° of the temperature of the condensing water. The importance of this is not relatively great because after it is once liquefied there is no more latent heat to remove; but if we take the specific heat of liquid ammonia at 1.2 even then six units would be removed from every pound of liquid passing for a reduction in temperature of only 5°.

A point established by the steam condenser experiments is the great superiority of horizontal as compared with vertical tubes and the importance of velocity in the movement of the condensing water. Makers of vertical tubular ammonia condensers appear to be very few in number, and results from their practice would be very interesting for comparison if exact tests had been made and were available.

The following table shows the effective surface of standard pipe used in the construction of condensers and refrigerating coils:

Inside Diam.	Outside Diam. in inches.	External Circumference in inches.	Length requir'd for a sq. ft.	Surface in sq. ft. of 1 ft. in length.
1	1.315	4.134	2.903	.344
1¼	1.66	5.215	2.301	.434
1½	1.90	5.969	2.201	.497
2	2.375	7.461	1.611	.612
2½	2.875	9.032	1.382	.752
3	3.50	10.966	1.091	.911
3¼	4.0	12.566	0.955	1.074
4	4.5	14.137	0.849	1.178

The following table shows the number of thermal units to be abstracted to be equivalent to one ton refrigeration in twenty-four hours:

	Per Day.	Per Hour.	Per Minute.
American Ton	284,000	11,833	197.2
English Ton	312,080	26,060	216.7

From the information contained in the preceding tables it is possible to calculate the length of pipe required for any given amount of refrigeration when the temperatures of the two substances on the inside and out are known or assumed. As a practical supplement to this part of the whole refrigera-

tion question, the actual proportions of a number of condensers by different makers have been collected and are given in tabular form for easy reference and comparison, as follows:

Different condensers.	Lineal feet of pipe.	Size of pipe.	Superficial feet per ton ice making.	Superficial feet per ton refrigeration (or equivalent).
ATMOSPHERIC CONDENSERS.				
*"Antarctic," Sydney	218	1¼	94.6	(47.3)
* " India...	133 133	1½ 1¼	66.1 57.7 123.8	(61.9)
Buffalo Co., specially made for Australia.	75	1½		37.2
*"Hercules," Sydney.	114	1¼		49.4
*"De La Vergne"...	40	2		24.8
*"Frick" (proposed).	62	1¼		27.1
"Consolidated" (from printed reports)....	100	1		34.0
As recommended in "Compend of Mechanical Refrigeration."............	115	1¼	99.8	49.9
From experience of E. T. Skinkle...	58 to 160	2		36 to 99.
Preference of E. T. Skinkle............	150	1		51.6
E. T. Skinkle, average of four plants from 25 to 100 tons, tabled by E. T. Skinkle.................	142	1		48.8
Average of three plants from 75 to 150 tons, tabled by E. T. Skinkle.................	99	1¼		42.9
SUBMERGED CONDENSERS.				
Recommended by E. T. Skinkle............	100	1		34.4
Recorded by E. T. Skinkle as average of eight machines from 10 to 140 tons...	89	1		30.6
PIPE REQUIRED IN FREEZING TANKS.				
Average of twelve plants from 2 to 60 tons. (E. T. Skinkle).	327 272	1 1¼	112. 118.	
"Consolidated," from records as printed..	320	1	110.	
"Antarctic," Sydney.	292	1¼	126.	

*Machines made for Australian use. Other authorities borrowed.

CHAPTER XIII.

COCKS, VALVES, PIPES AND JOINTS.

COCKS VERSUS VALVES.

Some makers pride themselves on the construction of their cocks, while others are thankful that, unlike their neighbors, they use nothing but valves. Every refrigeration plant requires cocks or stop-valves in great numbers besides the most important one which controls the connection from the condenser to the refrigerator and constitutes the last of the four principal features of the whole plant. These cocks or valves are required for both the forward and

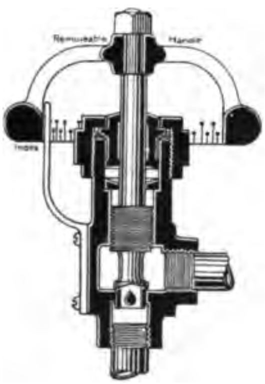


FIG. 26.

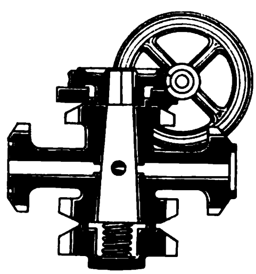


FIG. 27.

FIG. 28.

back pressures and much importance is attached to their construction.

During the early days of refrigeration, cocks were generally adopted which were made of cast iron with steel plugs, and most careful workmanship was required to secure a perfectly tight job. A similar arrangement is still used by some leading makers, but the preference on the whole seems at present to be given to the use of valves. Of the many well

known patterns now made for regulating the supply of ammonia to the refrigerator the most notable perhaps are the Frick valve, as Fig. 26, and the De La Vergne cock, Fig. 27. The most simple and reliable arrangement of expansion device for ordinary purposes, however, appears to be a hard steel valve with a long taper in a casing of iron or steel, as Fig. 28. (For latest Frick valve see Chapter XX.)

In Australia ammonia valves were formerly made from a solid block of hammered steel, and were in fact a well known

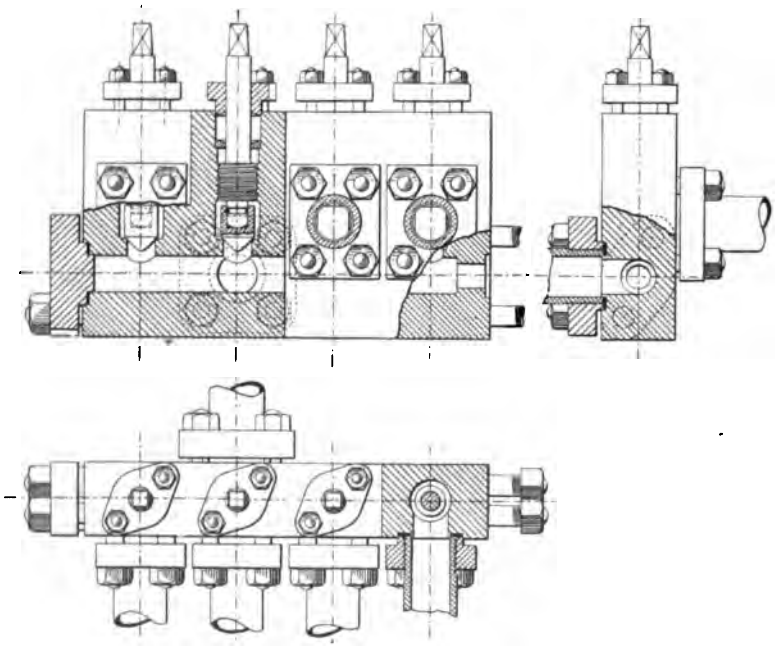


FIG. 29.—FORGED STEEL AMMONIA MANIFOLD VALVE.

hydraulic fitting modified to adapt it for ammonia. One of these—as made in manifold for connecting up the return ends to four coils in a refrigerator—is shown by Fig. 29. Such valves are of course more expensive to make than those with cast or malleable cast shells, but on account of their intrinsic merits they were largely adopted in high class work. Fig. 30 shows a solid steel main stop-valve with a by-pass valve, as used for the inlet and outlet of the compressor.

PIPES AND JOINTS.

The four principal factors in the constitution of a refrigerating plant so far referred to would be useless if they were not connected together by conduits or pipes. Owing to the

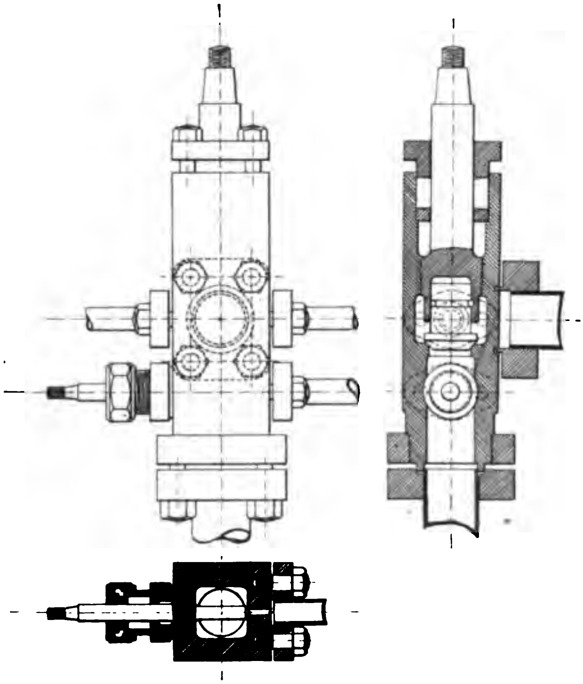


FIG. 30.—FORGED STEEL MAIN VALVE WITH BY-PASS.

action of ammonia on copper and its alloys, as already referred to, iron or steel must be employed for ammonia fittings. Lap-welded tubes are preferred to cast iron pipes

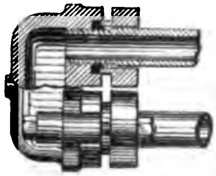


FIG. 31.

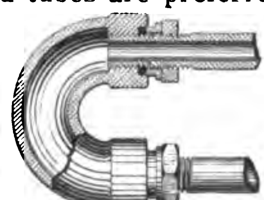


FIG. 32.

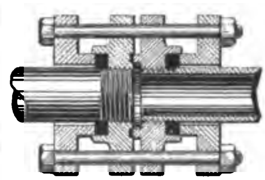
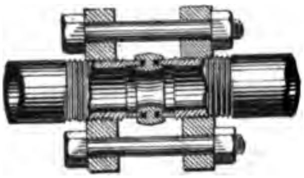
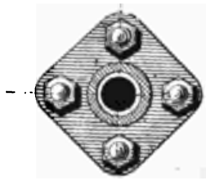


FIG. 33.

for this purpose owing to the risk of leakage through porosity or sponginess in the castings. Mention has before been made of the necessity for absolutely tight joints, and great

ingenuity has been expended in devising every conceivable arrangement of joint possible for connecting the separate lengths of wrought iron pipes, often no doubt in order that makers might either have a patent or a claim for a system





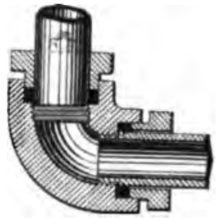


FIG. 34.—HUDSON'S PATENT. FIG. 35. FIG. 36.

of their own. Figs. 31 to 43 show a number of these devices which almost explain themselves; some of them have been invented and patented more than once.*

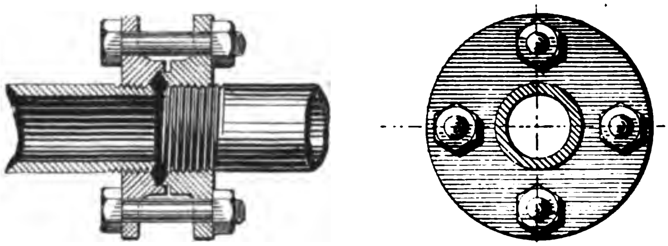
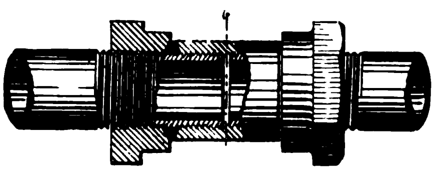


FIG. 37.—AULDJO'S PATENT JOINT.

More loss and trouble are often caused by cheap joints than would pay for the highest class of fittings in the first instance, and for a refrigerating plant, it is safe to say, noth-



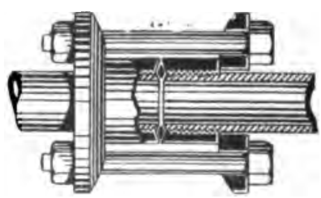


FIG. 38. AULDJO'S PATENT OTHER FORMS. FIG. 39.

ing is likely to be so dear as so-called cheap joints. Perhaps the very best all-round joint yet introduced for welded tubes,

*The joint, Fig. 40, has recently been patented in New South Wales by Douglas Kyle. It was invented years ago by the late David Boyle, and known as the Boyle joint, being patented in 1876. But strange to say it was illustrated in the German *Der Constructur* in 1868.

though not the cheapest in first cost, is that shown by Figs. 42 and 43, where the pipe is secured to the flange by sweating it with solder, as well as the screw thread, and the flanges are tongued and grooved together.

This joint has been made in Sydney for over thirty years, having been introduced by Mr. E. D. Nicolle, and has

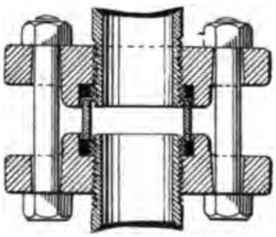


FIG. 40.

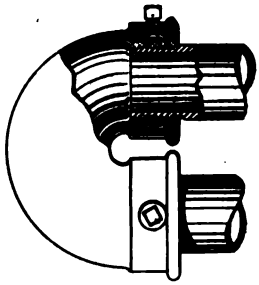
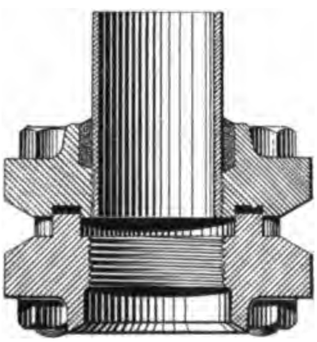


FIG. 41.

since been found to be the best by very large American builders of refrigerating machinery, who adopt it as their own, with a slightly modified shape of flange, but with the same male and female joint and recess for a metallic or other grommet.



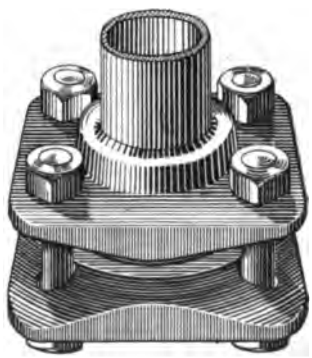


FIG. 42. AN AUSTRALIAN AMMONIA JOINT. FIG. 43.

ELECTRIC WELDING.

The introduction of electric welding by which pipes can now be made up into long continuous coils has been a great boon to makers of refrigerating machinery, and has enabled joints to be largely dispensed with in out-of-the-way places, where a leak would be difficult to detect and stop.

(6)

In shops where the amount of coil work turned out does not warrant the outlay for an electric welding plant, wrought iron pipes of good quality may be successfully welded in an ordinary fire, after the two ends have been machined so as to make a male and female cone. A special mandril should be introduced during the swaging. Long lengths so treated, and afterward bent on the welds have stood the test pressure of 1,500 pounds per square inch, as well as electrically welded tubes.

SEVERAL DESCRIPTIONS OF COILS EMPLOYED.

When all the separate lengths of tube required for one section of a refrigerator or condenser are welded up into a continuous length they can be easily bent into the kind of coil required, whether a plain helix or spiral, as in Figs. 18, 21 and 22, or an oblong spiral, as in Fig. 23, and the several turns of the coils can be laid as closely together vertically as desired; but the horizontal distance of the two sides apart must be greater, being regulated by the radius of the bends at the ends. When, however, a zigzag arrangement with vertical returns is desired, then the several lengths must be spaced wider apart vertically on account of these bends in the tube; and in order to get the greatest number of lengths in the space available the inclined arrangement, as shown in Fig. 25, is often adopted. This design is in some respects objectionable because the liquid must be all evaporated in the first length and bend unless the pressure is sufficient to drive it up-hill to the next bend. The same objection does not apply to these coils laid horizontally, and condensers are sometimes made with vertical headers for the main inlet and outlet, connected by a number of zigzag coils placed horizontally one over the other, with or without valves. These condensers have the advantage of giving a short run, a large sectional area of passage and a slow velocity for the gas, hence they cause very little increase of pressure by friction.

When a zigzag coil is made of straight tubes and separate returns, as in Fig. 19, instead of with the bends in the tubes themselves, a condenser or refrigerator with a given number of lengths above one another can be kept much lower than otherwise, because the return ends may be cast

much closer than the wrought pipes could safely be bent to. Zigzag coils made in both ways are much used for the floors and sides of refrigerating chambers, those made with the bends in the pipe itself requiring much fewer connections. When built up from separate lengths they are generally connected at the returns in one or other of the following ways:

1. Cast metal returns, and the screwed and soldered male and female flanges with metallic grommet, as in Fig. 31, all connected up by bolts.

2. Cast metal returns, with screwed socket and additional recess for packing, the ends of the pipes screwed hard into the sockets, and followed up by a packing ring and a gland running on the thread of the pipe, as shown in Fig. 32.

3. Similar to 2, with the ends of the tubes screwed into socket, but with the gland to compress the packing running on the plain body of the tube, and drawn up by two bolts, as shown in Fig. 31.

4. Cast returns screwed right and left-hand alternately and formed with a recess containing soft metal packing that can be closed up by a set screw, and the pipes screwed right and left handed at opposite ends, as shown in Fig. 41. (The author has no personal experience with this joint, but it is claimed as a great advantage that any pipe or return can be easily changed under this system, and the whole kept easily tight.)

5. But when the returns are bent on the separate tubes themselves, then the joints on the straight portion of the tubes may be made with any form of flange or socket as used in any other position. Figs. 34 and 35 show the joint patented and used by a large firm of Sydney engineers, the ends of the pipes being machined to fit into a double-grooved socket.

6. With continuously welded coils the connection to manifolds or headers is frequently made by an Australian flange, as shown in Figs. 42 and 43.

CHAPTER XIV.

THE USE OF OIL IN REFRIGERATING SYSTEMS.

SUPPLY OF OIL TO THE COMPRESSOR.

In the early days of ammonia compression, and before the accurate mechanical construction now possible and usual was put into such machines, compressors would not deliver so large a percentage of the cylinder's total volume as they do now. The pistons were, no doubt, not so accurately fitted that the ammonia itself would furnish all the lubrication required, and the clearance was excessive. With a view to the expulsion of the whole cylinder's contents a system of compression was adopted for ammonia similar to that used with wet air compressors, and in some very high class machines now made the cylinder at every stroke receives an injection of liquid; and as this requires a substance which will not saponify under the action of ammonia, special grades of hydrocarbon or mineral oil are prepared for the purpose.

The advocates of such an arrangement contend that the oil not only fills all the interstices resulting from bad design, and reduces the effective clearance to *nil*, however great the mechanical clearance may be, but that it takes up a great deal of heat from the gas; and thus by reducing the volume of the same reduces the power required for the work of compression. No doubt all this is true in a degree, but it is at the expense of a reduced piston speed and therefore a reduced compressor capacity, because oil cannot be banged about as gas may be. Besides this, the plant must be provided with a complete system of pumps, separators and condensers for circulating and cooling such oil and restoring it to a reservoir, freed from ammonia, to be used over again. All of these special features have to be taken into account when comparing the first cost of plant and working expenses under this system with the cost of equal results obtained

from others. An inspection of Fig. 20, and comparison of the same with Figs. 17 and 24, will enable the much greater complexity of the oil system to be better understood.

NO OIL NECESSARY IN SOME COMPRESSORS.

Some makers of high class modern machinery claim that no oil at all is required for the pistons of their compressors, and only use it as a seal to the piston rod. In single-acting vertical compressors a little oil lying in the bottom of the cylinder around the neck bush must necessarily prevent the passage of gas through the packing, and it is only subjected in such cases to the back or expansion pressure. In ordinary double-acting compressors, however, the piston rod and its packing are subjected to the full forward or condensed pressure. In an ordinary double-acting compressor working horizontally, as in the Linde system, no body of oil can lie round the neck ring, and it is usual in such cases to have a very long stuffing-box, with a lantern bush separating two sets of packing, as shown by Fig. 44. A small oil pump, generally driven by the machine, or a lubricator, keeps up a supply of oil to this intermediate space, and under such an arrangement any ammonia that escapes is absorbed by the oil, which is carried to a special vessel, where it is separated and then used over again. A certain amount of oil is also carried on the piston rod into the cylinder to lubricate the piston at every stroke, which necessarily requires it more than a vertical machine. In vertical single-acting compressors an oil vessel is often attached which has a small hand-pump fitted to it by which the attendant can force oil into the bottom of the cylinder to seal the piston rod as required. See Figs. 45 and 46. The latter is a special design by the author, and has a glass bottom to show the quantity of oil in the well.

The oil which escapes through the packing in vertical compressors naturally runs down the piston rod, and to catch the same and keep the machine clean the piston rod often runs through a bowl or cup on the crosshead, as seen in Figs. 11 and 17.

In Fig. 11, where glycerine is used as a lubricant, which is forced in between the double leathers of the piston and

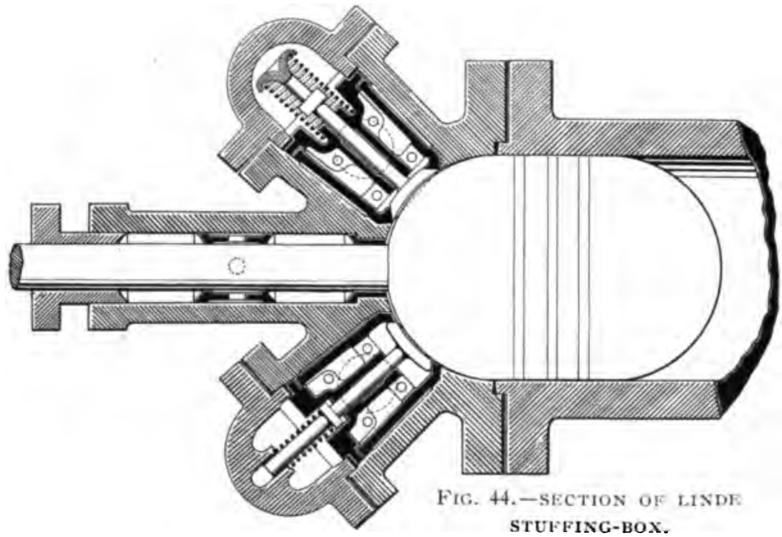


FIG. 44.—SECTION OF LINDE STUFFING-BOX.

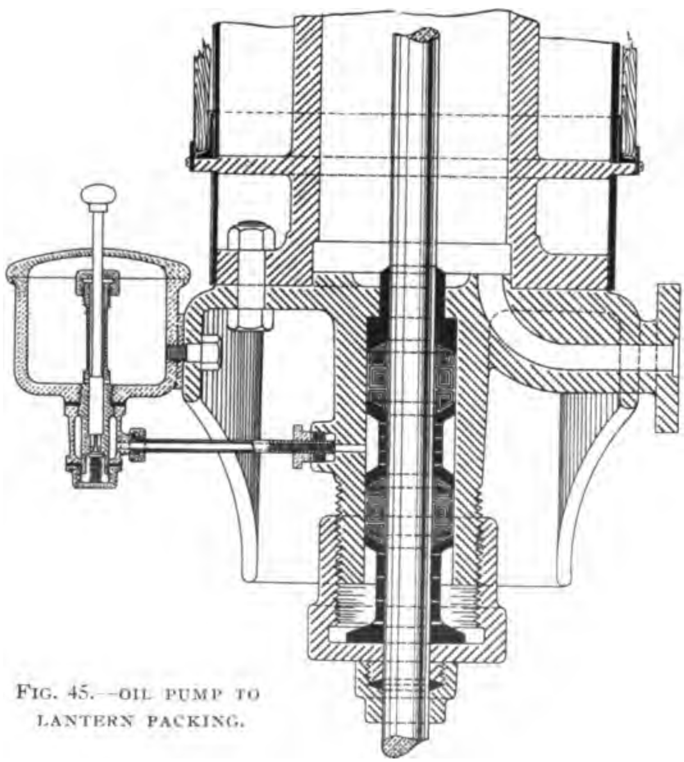


FIG. 45.—OIL PUMP TO LANTERN PACKING.

packing, the cup has an overflow pipe into a portable receiver, so arranged as to be emptied by hand. Fig. 47 shows a device specially designed by the author to intercept this oil by a second and lighter packing in a lower stuffing-box, and a circular trough with pipe to carry it to a receiver.

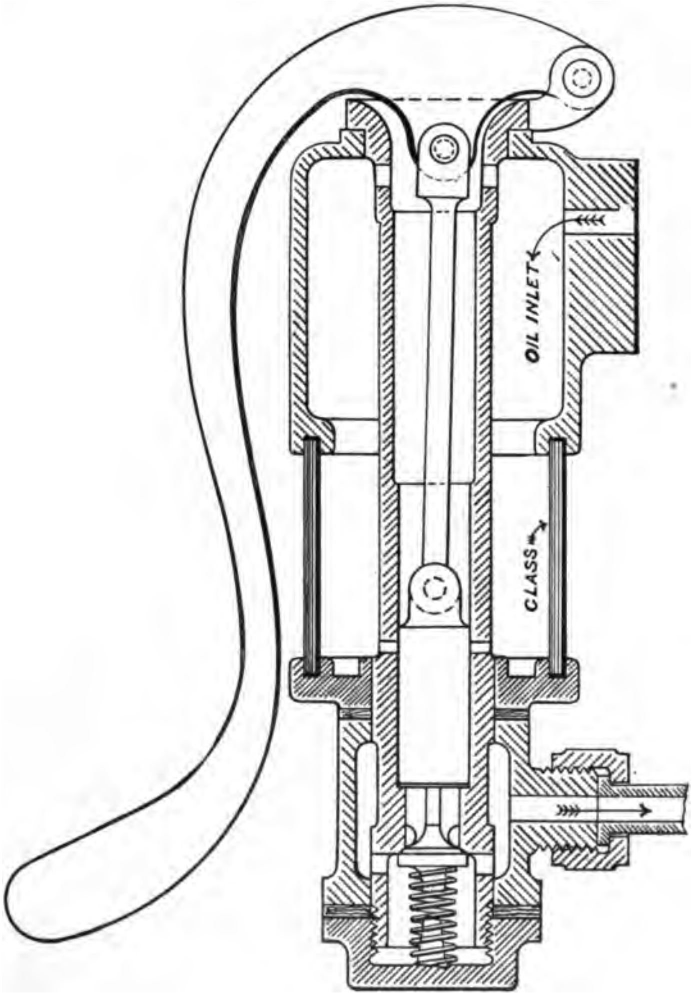


FIG. 46.—LUBRICATING PUMP WITH GLASS BODY.

It will be noticed that any oil which passes the upper or main packing can escape through openings above the lower or "swab" packing and run over a "drip" into an annular

channel; a pipe leads the oil from this channel to a reservoir, either cast in the frame or attached, whence it can run down to the glass reservoir of the pump seen in Fig. 46, to be again returned to the compressor.

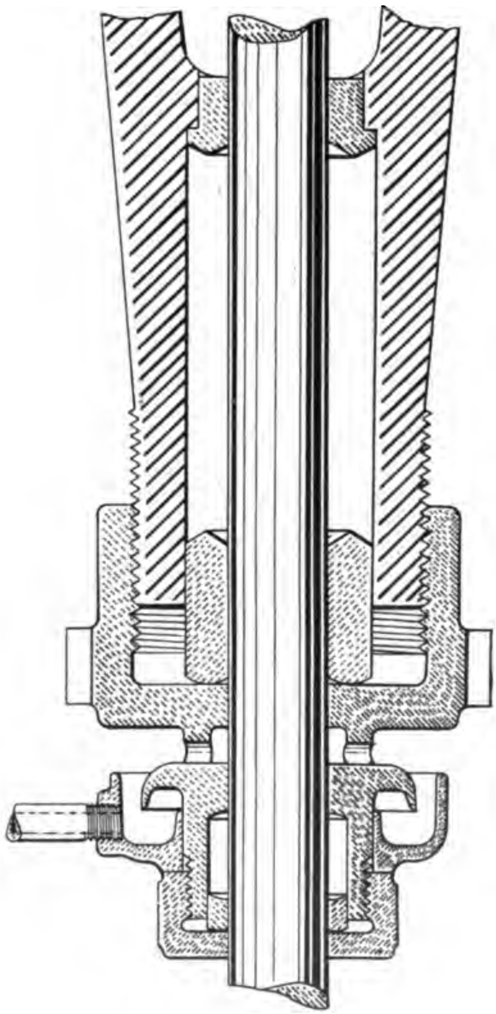


FIG. 47.—OIL INTERCEPTOR FOR PISTON ROD—BY THE AUTHOR.

In machines of the class shown by Fig. 11, where the pressure often runs up to 1,100 pounds to the inch, an extremely simple system of automatic lubrication of the

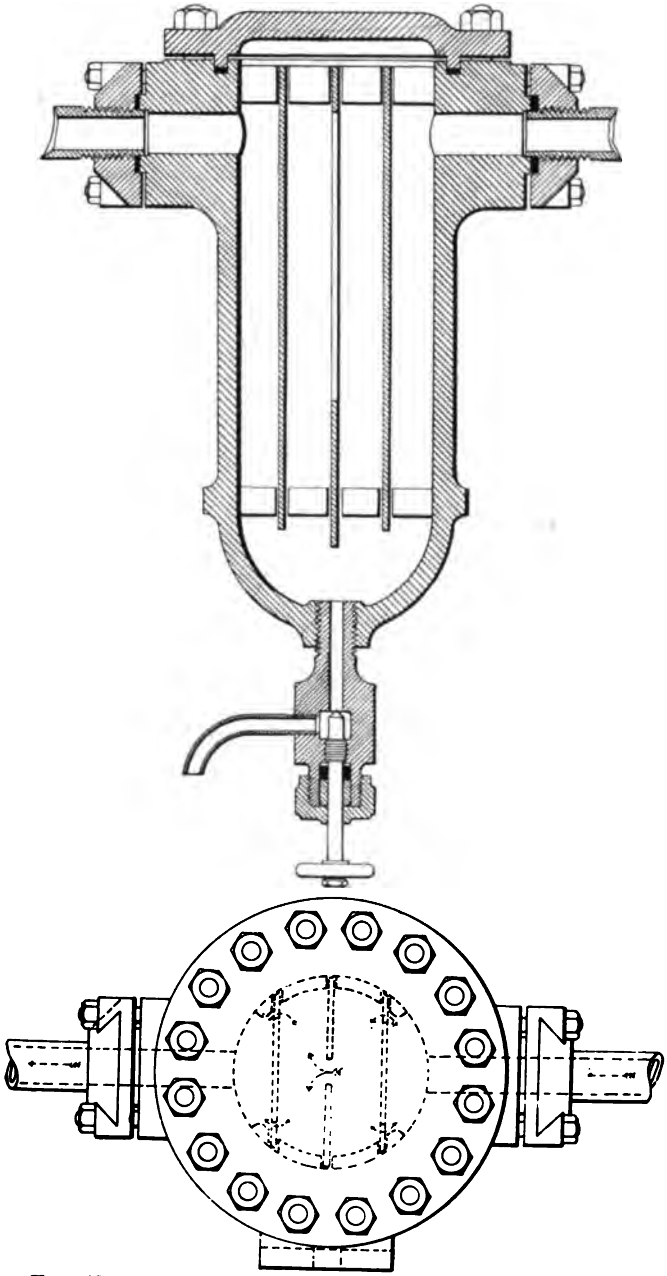


FIG. 48.—SECTION OF OIL SEPARATOR WITH "BAFFLES."

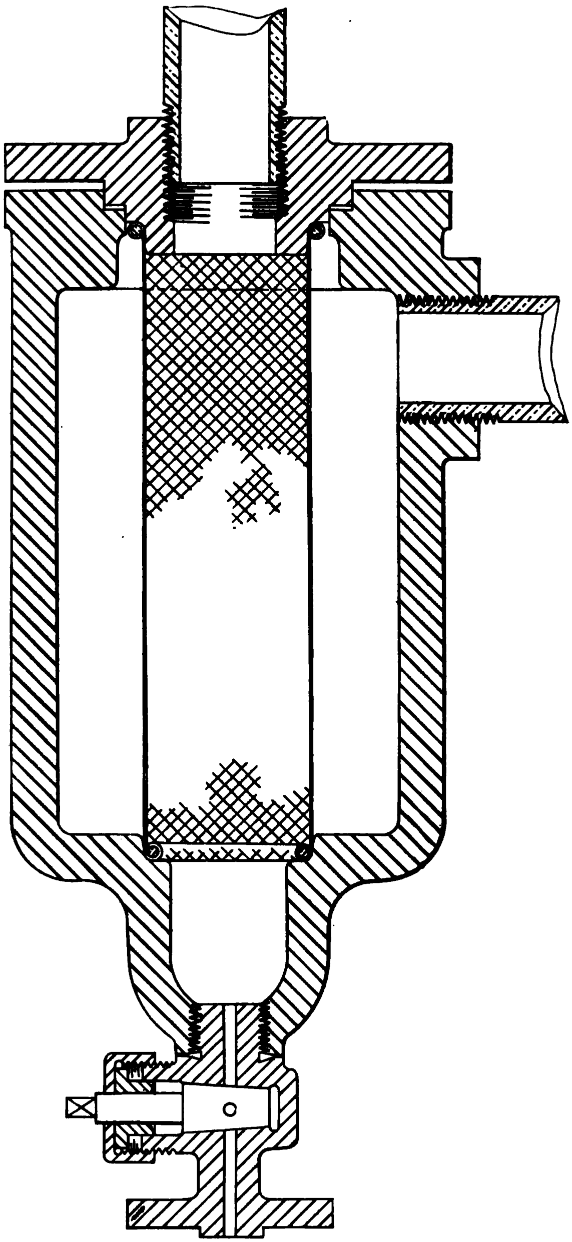


FIG. 49.—SECTION OF OIL SEPARATOR WITH WIRE SCREEN.

piston rod is adopted. The vessel shown at the side of the machine to hold the lubricant has a small pipe with regulating valve to adjust the flow to the packing, and also has a pipe which puts it in communication with the full forward pressure. After being filled with glycerine from the upper vessel the filling valve is closed and the pressure valve opened; it is then only necessary to adjust the small valve, seen on the pipe to the stuffing-box, to the flow required. Owing to the catches provided on the crosshead this can be used over and over again.

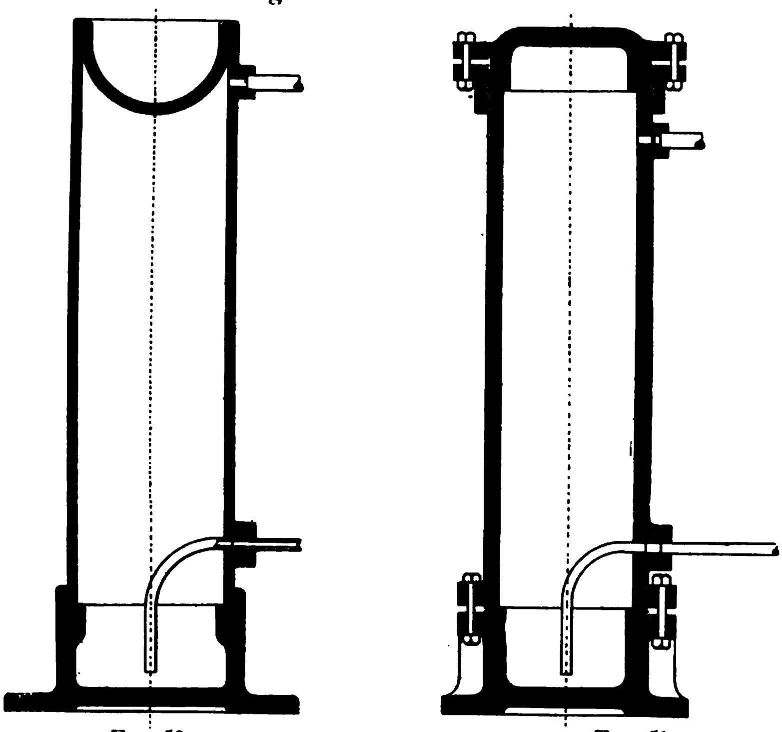


FIG. 50. LIQUID AMMONIA RECEIVERS. FIG. 51.

As an escape of gas takes place every time the lubricating vessel has to be filled, this system is not so well adapted for ammonia machines, but the small quantity of carbonic acid which escapes would not be noticed.

SEPARATION OF OIL FROM THE AMMONIA.

Seeing that oil is almost invariably used in refrigerating compressors, it becomes necessary to interpose certain ves-

sels in the course of a refrigerator system to prevent it being carried into the pipe coils of the condenser and refrigerator, where it would materially reduce the efficiency of the pipe surface as a conductor of heat. The principal oil separator in a system is usually fitted on the main pipe between

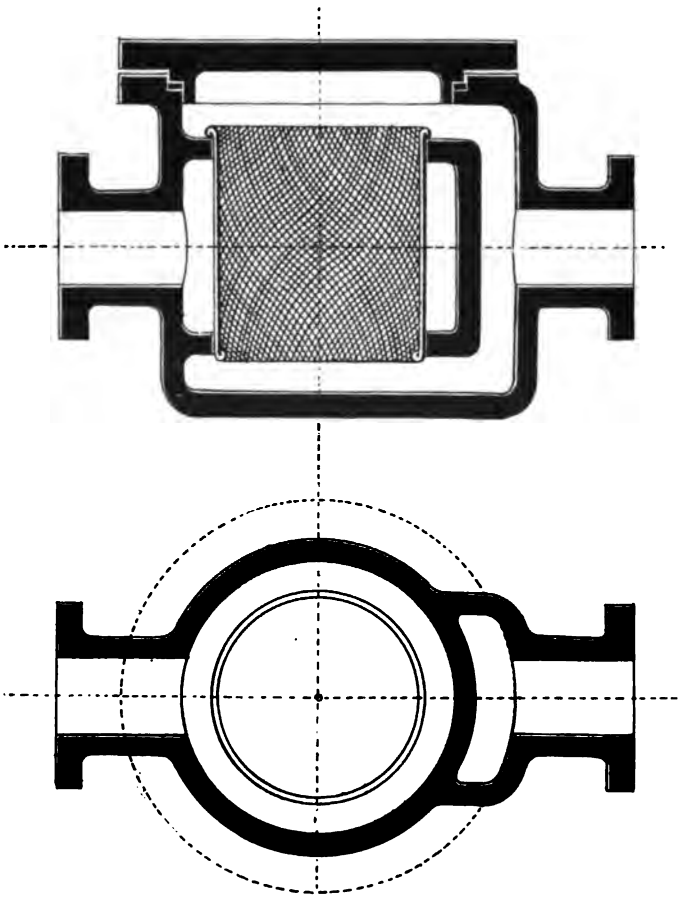


FIG. 52.—SECTION AND PLAN OF INTERCEPTOR.

the compressor and the condenser, and some experts attach great importance to this vessel being very large. Fig. 48 shows such a vessel as made for fixing to a wall, and provided with baffle plates to facilitate the deposition of the oil by the hot vapor. This deposition is facilitated if the vessel

is kept comparatively cool, which is difficult to do if it is too small. In some cases the outlet and inlet pipes to a separator are simply placed vertically through the top cover without anything to baffle or arrest the oil suspended in the hot vapor, and in others wire screens are introduced, as in Fig. 49. Opinions appear to differ greatly as to what is the best arrangement and proportion of parts for effectively keeping oil out of the condenser.

LIQUID AMMONIA RECEIVER.

Two separate forms of vessels for containing the liquid ammonia are shown by Figs. 50 and 51. This vessel is always placed below the condenser, and from it the supply pipe is led to the refrigerator, which is regulated by the expansion cock or valve, which is sometimes called the "flashing" or flash valve, the name no doubt suggested by the idea of liquid flashing into vapor as its pressure is removed when it passes into the refrigerator.

INTERCEPTOR OR TRAP.

Another vessel, to act as an interceptor or trap, is often placed on the expansion or low pressure side of the refrigerator, near to the inlet to the compressor, in order to intercept any foreign matter—such as scale or dirt—that may accumulate in or be carried from the pipes, and prevent the same from entering the cylinder of the compressor, where it might injure the piston or valves. All these vessels may be made and jointed in many ways so long as they are absolutely gas tight, but the general preference is for wrought iron or steel bodies welded up at one or both ends.

CHAPTER XV.

THE STEAM ENGINE AND THE COMPRESSOR —THEIR FUNCTIONS CONTRASTED.

In a steam engine high efficiency demands the production of a given power with the *minimum* weight of steam supplied from the boiler, but with a refrigerating plant high efficiency means passing the *maximum* weight of gas through the cylinder of the compressor with a given expenditure of power. Again, a steam boiler shows its efficiency by the evaporation of the *maximum* weight of water per pound of fuel burnt, while the efficiency of a refrigerator boiler or vaporizer is measured by the evaporation of the *minimum* weight of the liquid medium per unit of heat abstracted.

In the steam engine and the refrigerating machine the work done for a given expenditure of power is largely modified, and the efficiencies of both are discounted by disproportion of parts, clearance, leakage and friction ; thus, while the theories which are involved in the compression and expansion of gases and vapors are the same for everybody, yet the practical results attained with compressors, as with steam engines, differ widely, in accordance with the design and construction of the machines by their respective makers.

THE MECHANICAL OPERATION OF COMPRESSING A GAS.

In compressing any gas the design and construction of the compressor cylinder with its piston and valves is of very first importance, as they are the primary instruments concerned. The shafts, cranks, connecting rods, fly-wheels, steam cylinders or other portions of the prime movers which supply the power to the piston of such a cylinder, occupy, as accessories, a secondary though important part. Almost any form of compressing cylinder, good, bad or indifferent in

design or construction, may have its piston driven by almost any mechanical arrangement of cranks, rods or levers, also either ill or well designed, and may also receive its motion from steam, water or any other power, economical or wasteful, without at all affecting its quality or efficiency as a compressor.

It is therefore desirable in instituting a comparison between different types of refrigerating machinery to classify their various functions, so that they may be separately and properly compared, and the following appears to be a convenient division to adopt in considering the questions involved:

Firstly.—The construction of the compression pump itself, with its pistons and valves, and its efficiency for the work it has to do.

Secondly.—The connection between the motor piston of the engine and the driven piston of the compressor as affecting the simplicity and efficiency of the transfer of power from one to the other, and the first cost of the whole machine.

Thirdly.—The provision for minimizing wear and tear, reducing cost of maintenance, and simplifying access to working parts for inspection and repair.

THE QUALITIES THAT ARE DESIRABLE, OR THE CONDITIONS THAT SHOULD BE FULFILLED, IN AN IDEAL COMPRESSION MACHINE.

Under the first head just referred to may be placed the following characteristics, which are directly concerned with the work done on the gas:

1. On the in, or suction stroke, the cylinder should fill with gas at a pressure as little below that in the expansion coils as possible, and the outlet valve should be tight.

2. The piston and its rod should work with the maximum of tightness in order to prevent leakage, and with the minimum of friction, which (as it generates heat and requires extra power to overcome it) involves a two-fold loss.

3. On the out-stroke the inlet valve should not permit any leakage back, and the whole contents of the cylinder, less the minimum of clearance, should be discharged through the outlet valve at a pressure as little above that in the condenser as possible.

Under the second head: Dealing with the general design and construction of the whole, and noting that the very massive foundations which are required by some compressors and their steam engines must be taken into account when comparing the cost of the same in working order—

4. The machine—other things being equal—should be self-contained on one sole plate so as to be easily and cheaply erected on the minimum of necessary foundations.

Seeing that with single and double-acting cylinders of equal capacity and piston speed, single-acting machines must have double the piston area of double-acting ones, and therefore transmit double the stress to the connecting rods and cranks, then—

5. The work of the compressor with its crank, rods and crossheads should be double-acting instead of single-acting, and the ratio of compression should be as small as possible during both strokes, in order to distribute the work over as large a portion of the crank pin's path as possible.

If it is required to minimize the strain on the crank pins, shafts and connecting rods, and keep down the weight, cost, friction and wear of those parts, and high mechanical efficiency with low working expenses are aimed at, then—

6. In order to minimize the friction in the bearings and prevent the loss of power which results from indirect action the connection of the engine piston to the compressor piston should be as direct as possible, and the crank shaft with the crank pins and connecting rods should only be required to take up and transmit the *difference* between the power exerted by the steam and that required by compressor pistons, respectively, at any given position, instead of having to carry the work and friction due to the *sum* of those powers.

7. The pistons and valves should be easily accessible for examination and renewal.

Under the third head, and connected with the maintenance of the whole of machine in working order—

8. All covers or bonnets should be made with a simple joint, and to insure perfect absence of leakage, such things as double or treble connections, with bridges under one joint face, should be avoided.

Lastly, all wearing surfaces should be adjustable and easily adjusted.

THE RESISTANCE TO A COMPRESSOR PISTON IS NOT UNIFORM THROUGHOUT THE WHOLE STROKE.

The curves in Fig. 5 show how the pressure in a cylinder increases as air or gas is compressed and its volume reduced. Leaving for the present the question of the difference between adiabatic and isothermal lines, it may be assumed that in practice, the actual curve of compression is always somewhere between the two, and that such curve can be ascertained at any time when a compressor is fitted with a suitable indicator. This instrument takes a diagram which shows the work done *by* the piston of a compressor, just as a diagram from a steam cylinder shows the work done *on* the piston of an engine. An engine piston commences its stroke with the *maximum* pressure acting upon it, which continues until the steam is shut off, when the force or power of the same diminishes by the ratio of expansion to the end of its travel; but the piston of a compressor commences its stroke with the *minimum* of resistance, or without having any resistance to meet at all apart from friction, because the gas is then, or should be, of equal pressure on both sides of it.

The resistance to the piston, however, commences with its movement, and the pressure of the gas in front rises until the condenser pressure is reached, and then it continues uniform as it passes the outlet valve to the end of the stroke. It is not all expelled, however, in practice, because a certain amount, more or less, is left in the space between the piston and cylinder head, called the "clearance."

Now this question of clearance has been the *bête-noir* or bugbear of generations of compressor builders, and its importance is sometimes forcibly brought home to machine men when they see a cylinder head fly clear of the studs through having too little clearance. In other cases a very small effective result is obtained through the machine having too much clearance.

It is easy to understand that as the ratio of compression becomes greater, so much the shorter is the latter part of the stroke during which actual delivery of gas takes place;

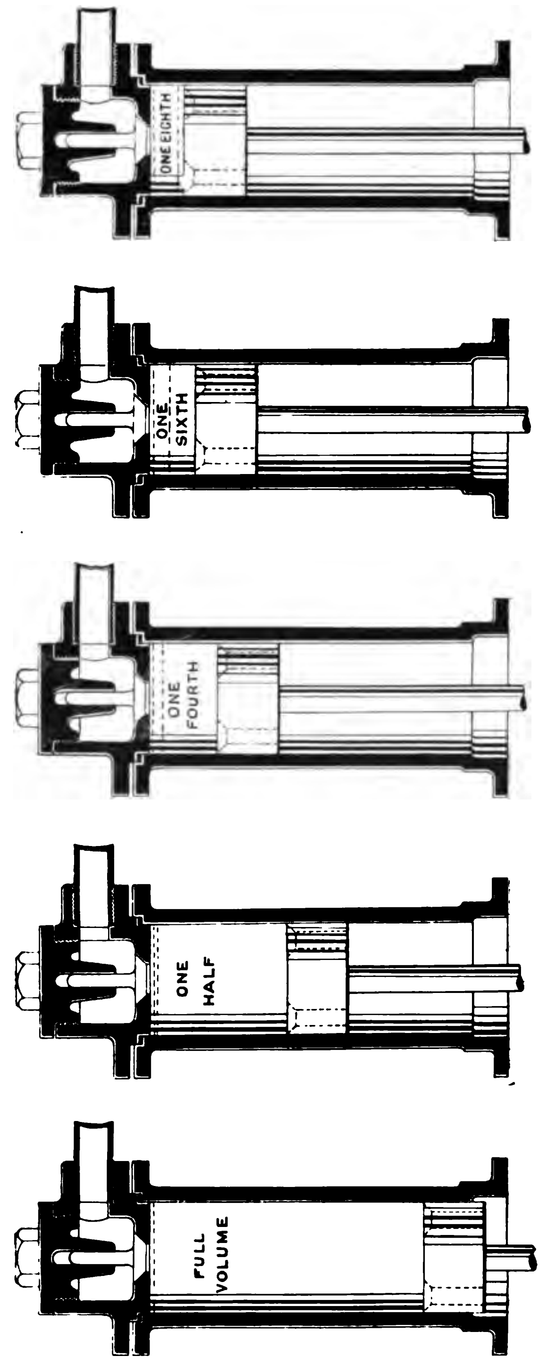


FIG. 53.—DIAGRAM TO ILLUSTRATE THE EFFECT OF "CLEARANCE" WITH DIFFERENT RATIOS OF COMPRESSION.

and, therefore, the greater the ratio of compression the greater is the loss with a given amount of clearance.

Fig. 53 shows five diagrams of a compressor, each one with the piston in a different position. In the first one the piston is at the bottom and before compression commences, and the cylinder is supposed to be full at normal pressure; the others show the respective positions at which the piston arrives before

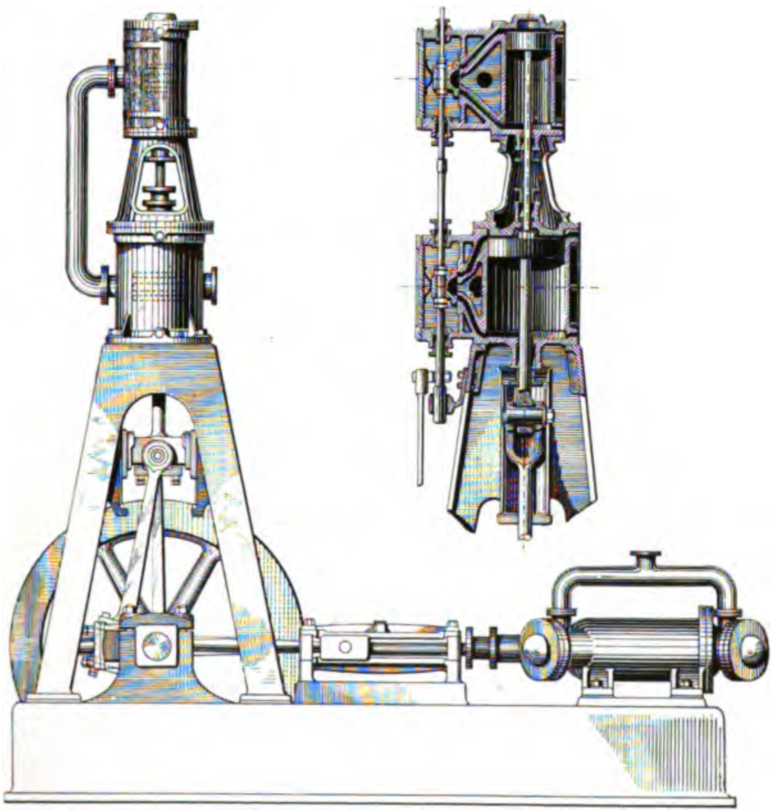


FIG. 54.—COMPOUND TANDEM ENGINE AND COMPRESSOR.
Designed by the author in 1881.

the gas is compressed into one-half, one fourth, one-sixth or one-eighth of its original volume; or, if it is an air compressor, then to two, four, six or eight atmospheres respectively. (The effect of the heat of compression is omitted in all these cases.)

The whole of the parallelogram between the piston and cylinder head in each instance represents the volume

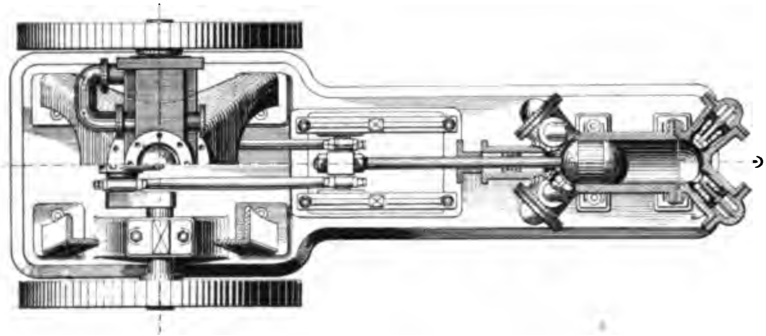


FIG. 55.—PLAN OF 1881 MACHINE BY THE AUTHOR.

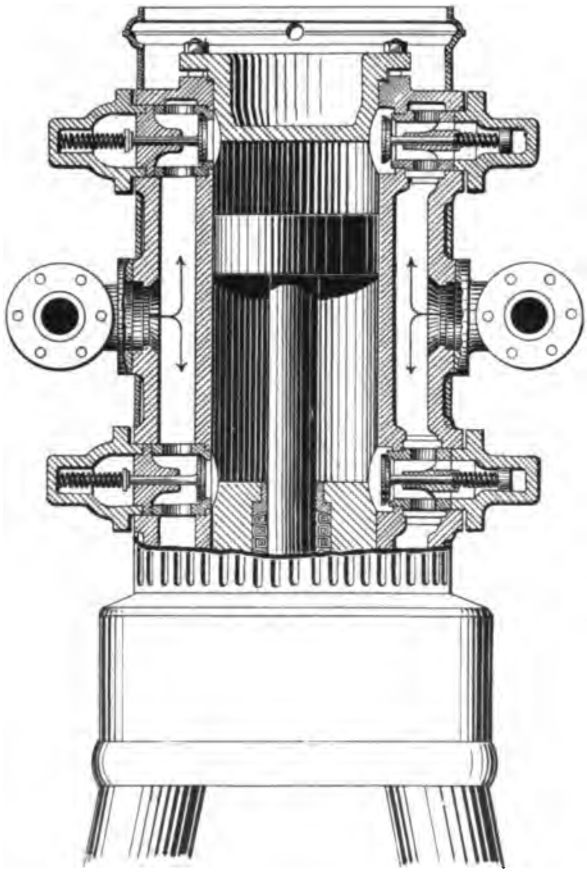


FIG. 56.—SECTION OF CASE COMPRESSOR, BUFFALO, N. Y.

at the increased pressure which would be delivered through the outlet valve if the piston was to strike the cylinder head at the end of the stroke. The space between the head and the upper dotted line represents an amount of clearance equal in all cases. The space between the cylinder head and the lower dotted line represents the volume to which the enclosed gas would re-expand and the line to which the piston would return before the cylinder could commence to refill on the return stroke. If this clearance is as much as one-eighth of an inch, then the waste or lost spaces would be one-quarter, one-half, three-quarters and one inch respectively, which would be deducted from the effective

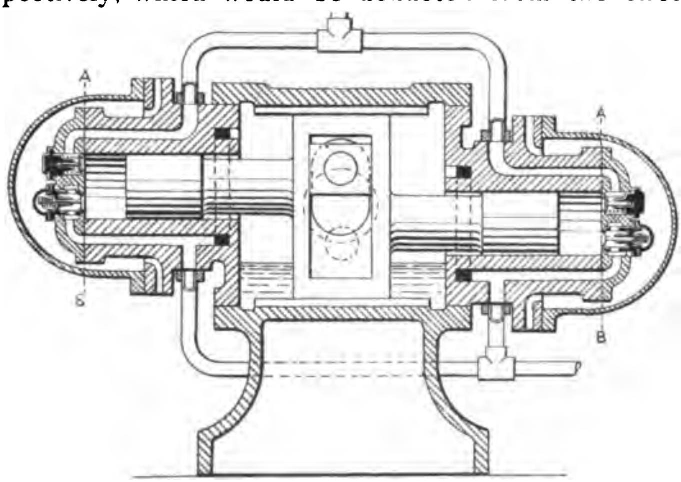


FIG. 57.—SECTION OF WESTINGHOUSE ENCLOSED COMPRESSOR.

stroke in the several cases. This shows that there would be a very large percentage of loss with high ratios of compression that would be intensified with short-stroke pistons.

This elementary explanation is no doubt unnecessary to many readers, but it paves the way for the proper consideration of the design and construction of compressors as actually built, and of their methods of meeting the conditions required for high efficiency.

SOME METHODS ADOPTED IN THE CONSTRUCTION OF REFRIGERATING COMPRESSORS TO MEET THE FOREGOING CONDITIONS.

In Figs. 54 to 70 there will be found sections of a number of compressor cylinders including well known and widely

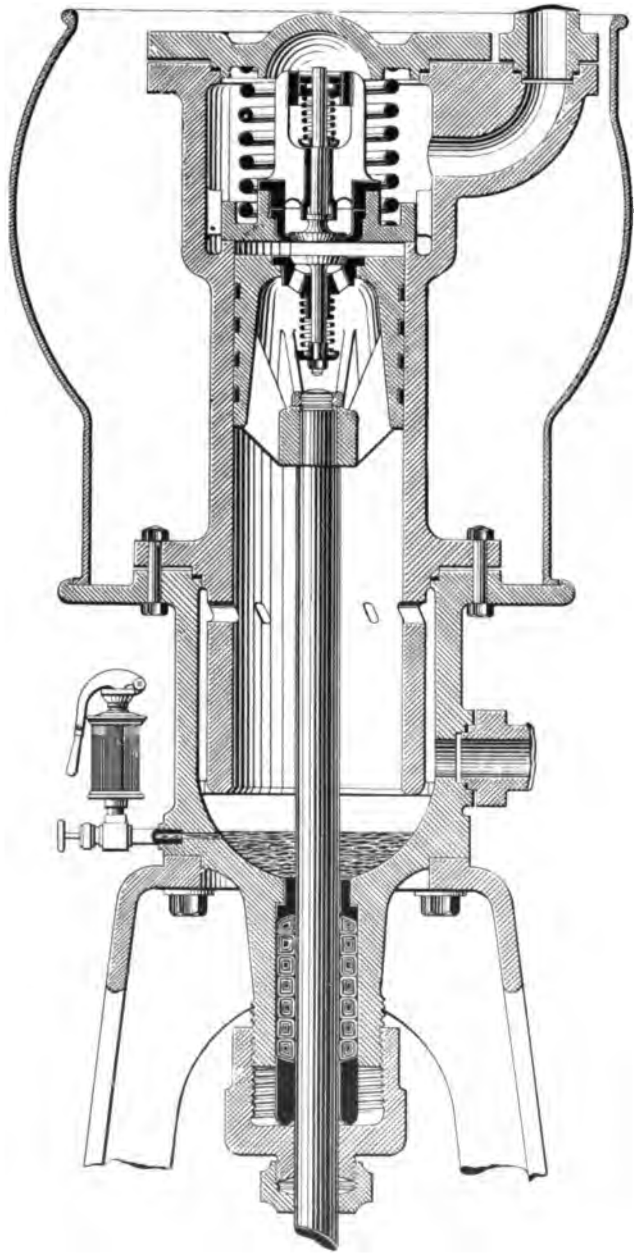


FIG. 58.—SECTION OF ANTARCTIC SINGLE-ACTING COMPRESSOR.
Designed by the author.

different types. An examination into their construction will enable us to see how they secure the several requirements which have been considered important in previous chapters.

FIRST.—*The cylinder should fill with gas as little below the pressure in the expansion coils as possible, or, in other words, exhaust the maximum weight from the refrigerator.*

Figs. 44 and 56 are sections of two double-acting compressors, the former working horizontally and the other vertically, but in both cases the inlet and delivery valves are placed with their axes lying horizontal. Such valves will of course not close by gravity alone. Fig. 57 represents a different type of compressor with two single-acting horizontal cylinders, and it also has horizontal valves. As such valves have no tendency to close by themselves, they require strong springs to insure their action being prompt and decisive, and therefore their cylinders never can fill to the full back pressure, because it is evident that during the admission of the gas there must always be a sufficient difference between the inside and outside pressure to overpower the resistance of the springs and open the inlet valves.

Figs. 54 and 55 show a compressor designed by the author some years ago with spherical ends to the cylinder and piston, so as to provide a larger area for the inlet and outlet valves; this is similar to the arrangement adopted in the well known Linde machines, Figs. 44 and 71, and seems to be the best possible arrangement for ordinary horizontal compressors with horizontal valves. Neither of these, however, can provide a perfectly free inlet for gas.

In Fig. 58, a design by the author (Sydney), Fig. 59, the Hercules (American), and Fig. 60, the Auldjo (Australian), all single-acting vertical compressors, it will be seen that special devices are in all cases provided whereby free communication is established between the inlet branch from their refrigerators and the interior of the cylinders when their pistons are right down. In Fig. 61—Antarctic compound —a similar arrangement is shown in the primary or low pressure cylinder.

In several of these compressors the pistons when on the bottom center uncover the ports shown, which open right through their cylinder walls, and in the case of the Auldjo

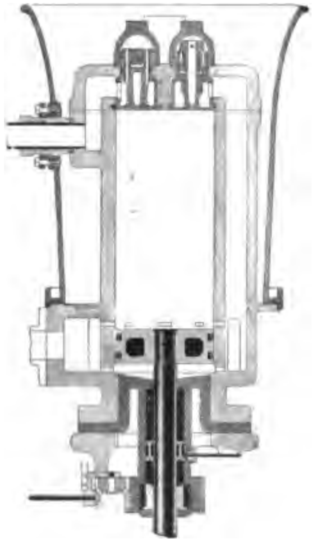


FIG. 59.—SECTION OF HERCULES COMPRESSOR.

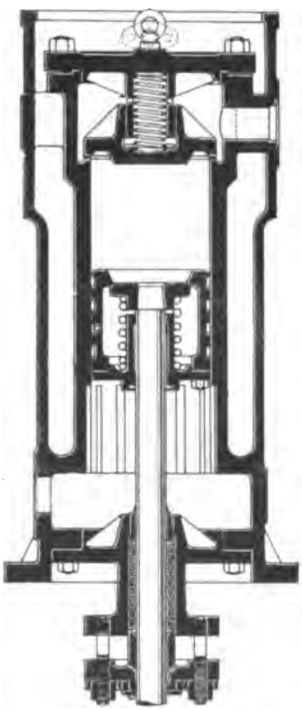


FIG. 60.—AULDJO COMPRESSOR.

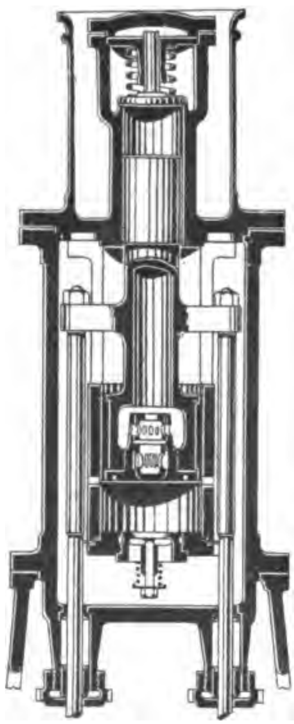
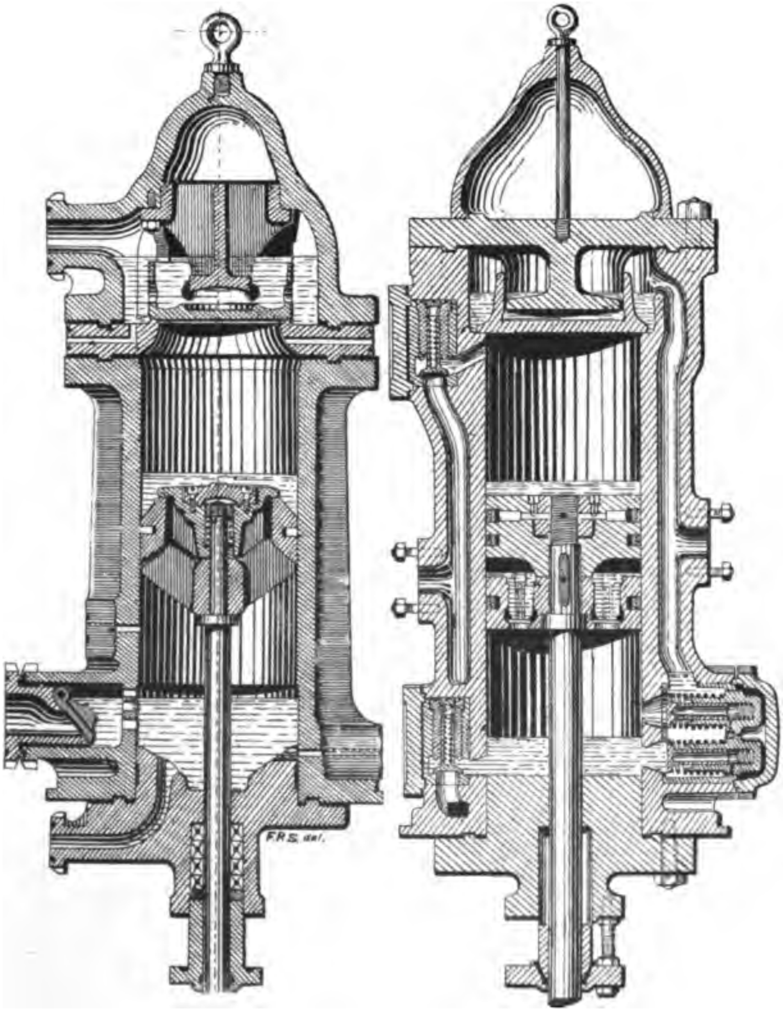


FIG. 61.—ANTARCTIC COMPRESSOR.

machine the piston passes the end of flutes or grooves cut in the walls of the cylinder. All of these cylinders can therefore fill with gas without any restriction, because an



DE LA VERGNE COMPRESSORS.
FIG. 62.—SINGLE-ACTING. FIG. 63.—DOUBLE-ACTING.

equilibrium is insured between the two sides of their pistons, whatever the pressure on the springs of the inlet valves may be. This idea, borrowed no doubt from the old fashioned air gun pumps, is supplemented in the Auldjo com-

pressor by an arrangement for opening the inlet valve automatically; this is effected by having the piston itself loose on the piston rod, and the valve itself fast on the rod in such a way that it opens on the down and closes on the up stroke. This makes a double (and what would almost appear to be an unnecessary) provision for securing a full cylinder of gas.

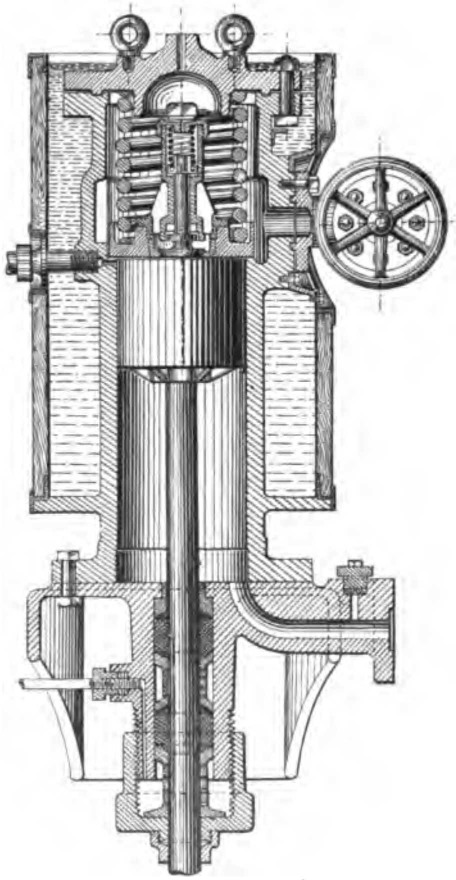


FIG. 64.—SECTION OF FRICK CO.'S COMPRESSOR.

In the cryogen machines—small ammonia dairy refrigerators, made in Queensland—and other small compressors there are no inlet valves at all, and the inlet is entirely provided for by the piston passing the end of grooves machined in the bottom part of the cylinder, as in the Auldjo compressor. In the Hercules machine there is a belt or passage

cast around the bottom of the cylinder which is in connection with the inlet branch, and into this belt holes are cored (not bored) through the walls of the barrel. Some of these holes

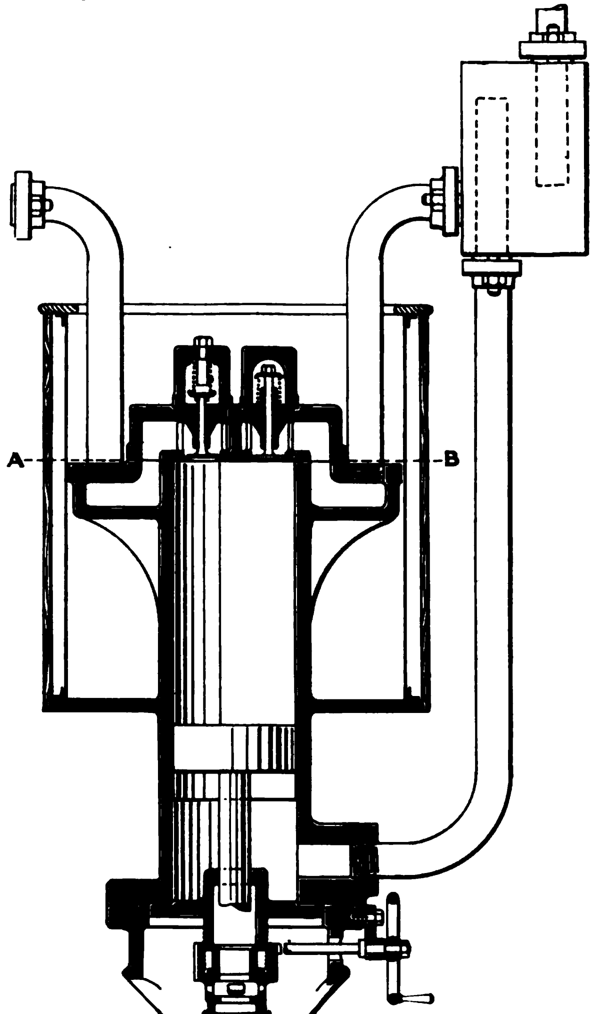


FIG. 65.—SECTION OF "CONSOLIDATED" COMPRESSOR.

are above and some are below the piston when it is down, and the gas has thus free access—quite apart from the valves—

before the return stroke. This arrangement involves a rather complicated cylinder casting, but the holes compensate for the necessarily restricted size of the inlet valve and secure the full back pressure of gas above the piston before compression is commenced.

In the compressors, Figs. 58 and 61, similar holes for admitting gas are provided, but instead of being cored, as in

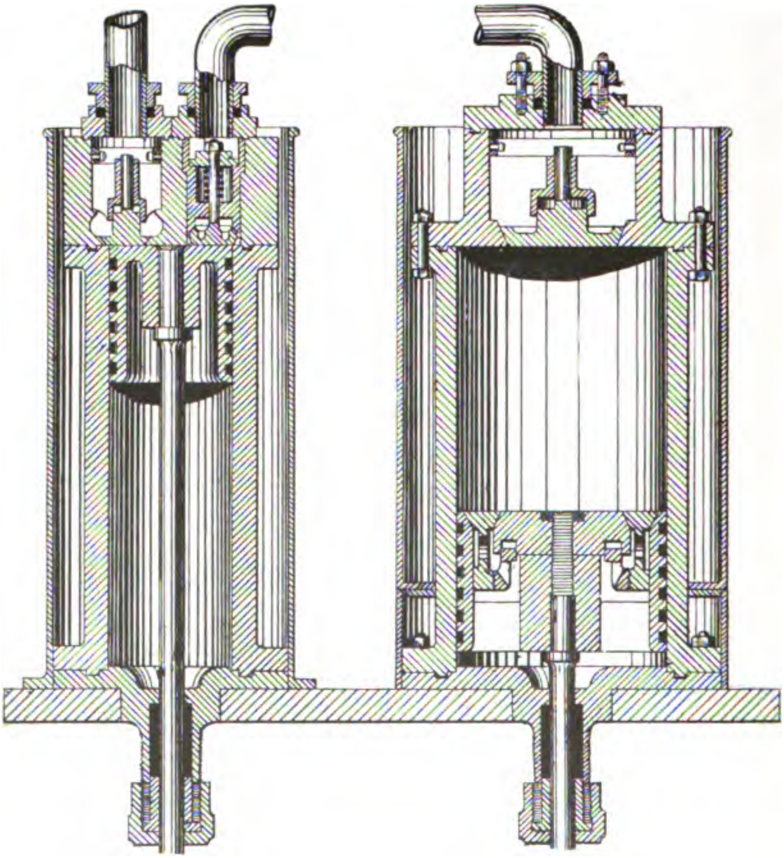


FIG. 66.—SECTION OF YORK CO.'S COMPOUND COMPRESSOR.

the previous case, they are drilled from the outside. This is an easy process with these machines because the working cylinders in both cases are made as plain barrel castings.

In the widely used De La Vergne compressors, Figs. 62 and 63, one of which is single-acting and the other double-

acting, the weight of oil would appear to affect the free admission of the gas, and the small valves in the double-acting piston of Fig. 63 probably reduce somewhat the effective pressure in the cylinder. As these machines run at a comparatively low piston speed, however, the actual loss may not be so serious as would otherwise be the case.

A broad contrast to the last example is seen in the Frick or "Eclipse" compressor, Fig. 64, which has the inlet valve in the piston made so large and so nicely balanced on springs, that when it has completed its down stroke there can be scarcely any difference between the pressure in the cylinder above and below the piston, and thus the filling of its cylinder is insured.

In Fig. 65, the "Consolidated" compressor, and Fig. 66, the York compound compressor, all the gas has to be drawn in through the suction valves, which have to share the space on the heads of their cylinders along with the delivery valves, and are thus restricted as to size. Looking at all these details and comparing their relative effects, it may be said that the *first condition* is more perfectly met (although by different methods) in such machines as the Auldjo, Antarctic, Hercules and Frick.

SECONDLY.—*The piston and rod should work gas-tight with the minimum of friction.*

With compressors, such as are shown by Figs. 62 and 63, the oil in the bottom of the cylinders must prevent any gas from escaping through the piston rod packing, although in the double-acting one it is subjected to the full forward pressure of the gas. The oil used in both these cases is intended to be carried right through the system very rapidly, in order to take up some of the heat of compression, and it is supplied at every stroke by means of a special pump. There is no need therefore for heavy packing and great friction in these machines. It is claimed for the Frick machine, Fig. 64, that specially good workmanship enables oil to be dispensed with altogether for lubricating the piston, except so far as it is carried in by the rod, and it is used only in a lantern bush, which is interposed between two separate packings in the stuffing-box, where it is forced in by a hand-pump. There must, however, be extra friction here, due to the exces-

sive length of the two packings, and as a matter of fact this lantern bush is not at all necessary for single-acting vertical types of compressors with accurate workmanship in the boring and turning of stuffing-box, glands and piston rods, while with the double-acting horizontal compressors, such as the Linde and those shown by Figs. 44 and 54, they are almost indispensable.

A pump driven by the engine is used in the Linde machines to eject the oil continuously between the two packings to prevent the escape of gas, and some of this is carried into the cylinder at every stroke. This of course does not apply to such Linde machines as are constructed with a lubricator on the stuffing-box instead of a pump. It will be noticed that in Fig. 58 the oil to seal the rod lies well below the inlet passage, and there is thus no tendency for the flow of gas to carry it up in quantity through the valve in the piston. In the compound compressor, Fig. 61, it will be further noticed that there are no piston rods proper passing into the cylinders at all, and that a depth of several inches of oil can lie in the bottom of the casing around the rods. In this case the tendency of the oil to pass through the system is minimized while full lubrication and sealing of the rods is secured.

In order that the piston of a compressor should work gas-tight, and yet with the least amount of friction and wear, it is imperative that the metal in the cylinder should be of a very hard and uniform texture. In order to better secure these qualities it is desirable that the cylinder itself should be made as a simple barrel or as plain a casting as possible. Any complication of cores, passages, flanges or projections upon a cylinder casting has a tendency to cause the metal to "draw" or become spongy, and make it very difficult to produce a sound, solid casting from specially hard iron. What is still worse perhaps is that an irregular casting has a tendency to alter its shape with every change of temperature, and as a compressor cylinder is subject to more changes of temperature than a steam cylinder, the desirability of having a casting that will be cylindrical at all temperatures and which will expand and contract equally all over is very evident.

This characteristic is most strongly shown in Figs. 57,

58 and 61, where the working cylinders are either separate bushes or quite plain barrels, and also in the Frick compressor, Fig. 64, where the working portion of the cylinder is quite plain. It is in a less degree in the Linde cylinder, which is generally made with the feet cast on. The most complicated cylinders to cast, owing to cores and passages, are probably the De La Vergne, Fig. 63, and the Hercules, Fig. 59, where the designs are such as to require great skill on the part of the molder to obtain sound and homogeneous castings which will wear uniformly all over. It will be noted that Fig. 61 represents a double-acting compressor in which the piston rod and its packing are never subjected to the forward pressure of the gas.

It must be within the knowledge of every one accustomed to compressors that cylinders often want reboring after a single season's work, and that pistons sometimes leak after being started only a few weeks, even if they were tight at first. The power of the engine has probably been employed to wear out the machine through undue friction. The remedy for this is to have cylinders made as plain castings of hard, homogeneous metal, accurately bored and lapped, and pistons that will work satisfactorily even if there are no rings in their grooves. Piston rings are extremely useful and necessary adjuncts, but, as often made, with a very strong spring to atone for a bad fitting piston, they are simply devices to wear out the cylinder and make the fit worse. An inspection and comparison of the several sections will show which are the types most likely to secure hard and absolutely sound castings.

THIRDLY.—*The whole contents of the cylinder, less the minimum deduction for clearance, should be discharged at the minimum of pressure.*

In the machines shown in section by Figs. 62 and 63 the presence of oil insures the full expulsion of the gas. In those shown in Figs. 58, 60 and 64, with movable heads to their cylinders, the pistons on the up-stroke may be so adjusted as absolutely to touch them, and thus the clearance is minimized in these cylinders. If smaller pilot valves are placed in the center of these valvular heads, then the compressor cylinders may be made as large as desired. In types, such

as those shown by Figs. 59 and 65, however, the size of the outlet valves is necessarily restricted, because there are two valves—both the inlet and the outlet—made in the one cover. These machines follow very closely in this feature the design

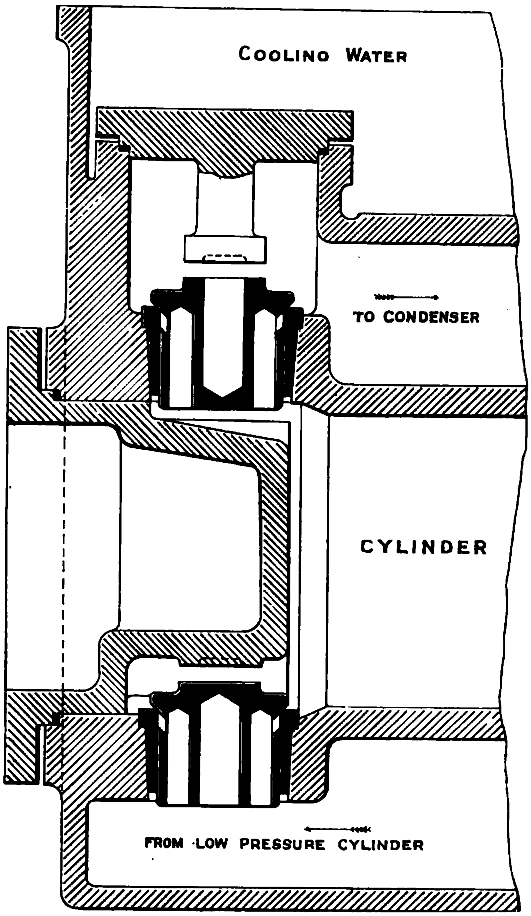


FIG. 67.—HIGH PRESSURE CYLINDER OF COMPOUND COMPRESSOR.

of some of the ether compressors of thirty-five years ago, and owing to such restriction in the delivery orifice require more clearance than is necessary for safety with larger outlet valves. The contracted size of the valves also increases the pressure to be overcome and reduces the piston speed.

These remarks apply in a modified way to the compressors shown in Figs. 54, 57 and 66.

The *pros* and *cons* of oil injection have been the subject of several interesting wordy wars which it is not necessary to touch upon here. Whatever he may once have thought of it, the writer does not now believe in the system. Apart, however, from the question whether the oil used in some of them absorbs and again gives out gas in their cylinders, the De La Vergne, Frick, Auldjo, Antarctic and others of that type are certainly the best fitted of all that have been so far illustrated for fulfilling this third function of fully expelling all the gas at the end of the stroke.

The amount of efficiency lost by a given amount of clearance in a compressor has already been shown to be dependent upon the ratio of compression carried out.

Thus one-sixteenth of an inch clearance with a two-fold compression would not cause so large a percentage of loss as one-thirty-second of an inch clearance with a five-fold compression in the same cylinder. It follows from this that when compression is carried out in stages, as in the "Lock" or St. Clair system, as in Fig. 66, or by the Antarctic system, as in Fig. 61, it is possible to get a very full discharge without a minimum of clearance; for let us suppose in a compound machine the high pressure cylinder to be only one-third of the area that a single compression one would require to be, then a given clearance in the same stroke would only waste one-third the volume otherwise lost. Fig. 67 is the back end of the high pressure cylinder of a compound compressor made for the author in 1884. It will be noted that, although the compressor is horizontal, the valves are vertical. Although the clearance is relatively large in this design, it is but of small comparative importance, as the ratio of second compression is only about 2 : 1.

In order to still further secure the maximum efficiency in preventing leakage past the pistons, the builders of some high class machines not only bore out their cylinders, but they lap them out afterward perfectly true, and then grind in their pistons. This is due to an advanced idea that the ordinary wear and leakage of cylinders and pistons is almost entirely due to defective material and workmanship, and that

the ideal piston that would never leak is the one that fits the cylinder so loosely as not to touch it, and yet so closely as not to permit the passage of gas. This is of course a question of workmanship; we know that a Whitworth gauge can be made so true that it cannot be passed through its collar with oil on it—as there is no room for oil—but will drop easily through it when dry polished with a silk handkerchief. In such a case there is evidence of good work. In the competition for business and the demand for cheap machinery of

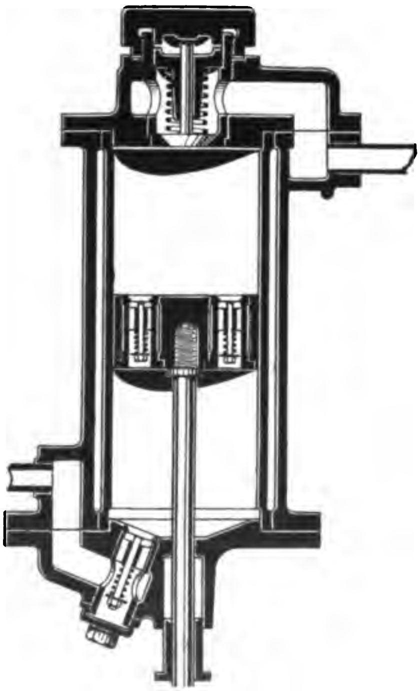


FIG. 68.—SINGLE-ACTING COMPRESSOR.
Patented in 1880 by the author.

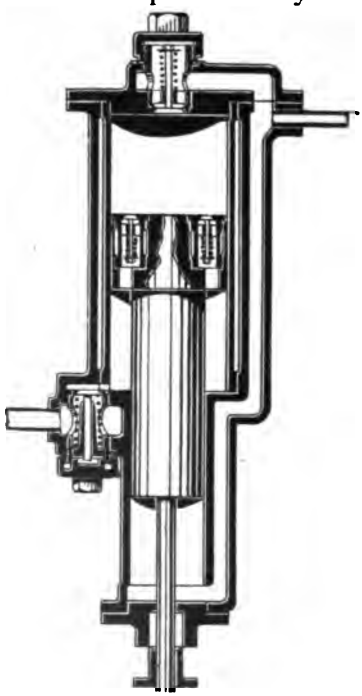


FIG. 69—COMPOUND COMPRESSOR.
Patented in 1880 by the author.

all kinds such high class work is perhaps not common in the construction of refrigerating compressors, and as a matter of fact, the best surfaces of ordinary piston and cylinder walls as they are left by the turning tools are like the ridges and furrows of a plowed field on a small scale, and they are often not so microscopic as to want more than an ordinary eye or finger to detect their inequalities. It is quite certain that a piston may be a very tight fit in a cylinder one day and

yet work easily enough to rattle about shortly afterward when the tops of the hills have been worn off the two metallic surfaces.

The author is an advocate for a true cylinder that will be equally true whether it is hot or cold, however much it may expand, and a piston which fits it and has such a thickness of metal as to heat and expand equally with the cylinder, and he does not like strong spring piston rings of hard steel, which are continually destroying good cylinders. It is better

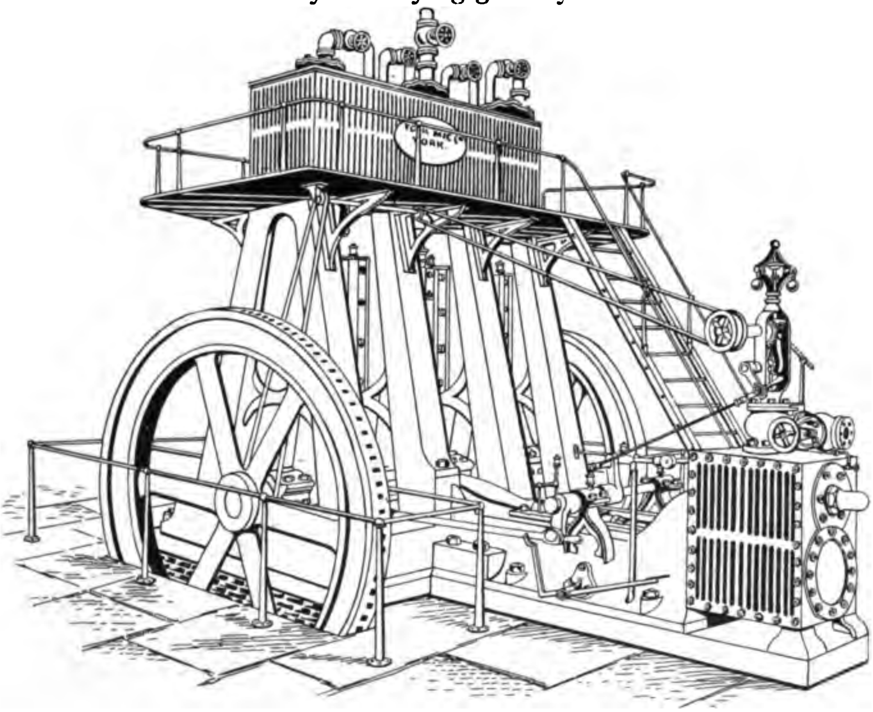


FIG. 70.—YORK CO.'S COMPOUND COMPRESSOR AND ENGINE.

to get a new piston than to spoil your cylinder with hard rings. To those who have never before seen an ammonia cylinder lapped out after being bored, it will come as a revelation when they first see it done and realize how imperfect is the surface of the ordinary cylinder that is turned out by the best lathe or boring mill alone.

Figs. 68 and 69 represent two designs, one of which is for a single-acting and the other for a compound ammonia

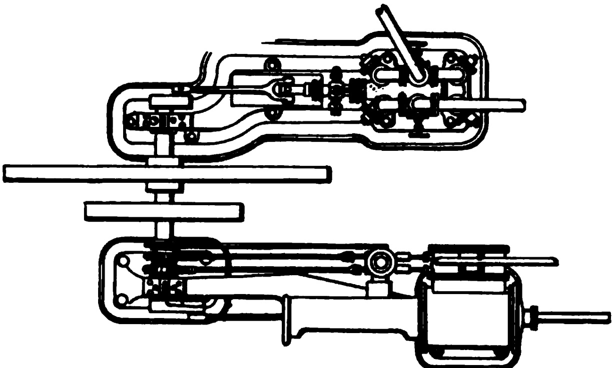


FIG. 71.—PLAN OF LINDE COMPRESSOR AND ENGINE.

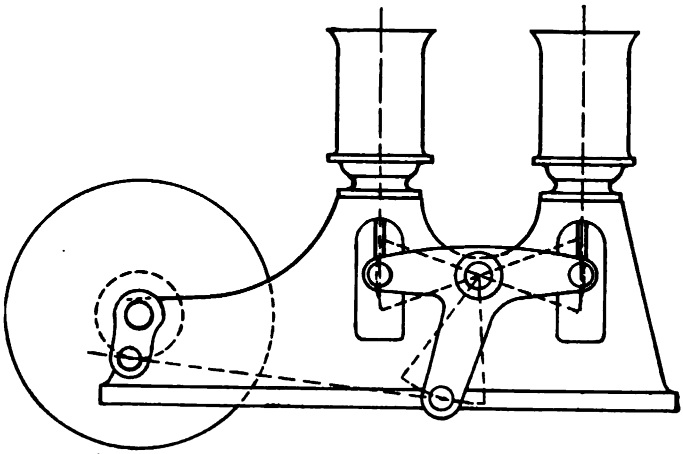


FIG. 72.—ELEVATION DIAGRAM OF HERCULES MACHINE.

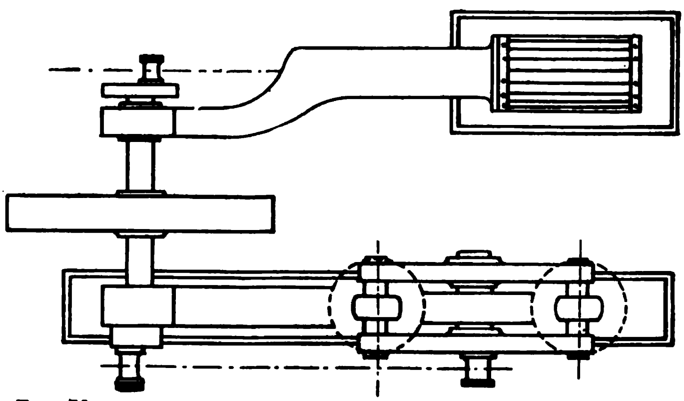


FIG. 73.—PLAN OF HERCULES MACHINE AND STEAM ENGINE.

compressor, which were patented by the author as far back as 1880. It will be noted that they both work with valves in their pistons, and that provision is made in both of them to insure that the piston rod packing is only subjected to the back pressure—an arrangement which has since come into very general use.

Before leaving the subject of compound compressing cylinders for the present, it will be well to note that, although it is not important in small machines and plants, yet with compound compressors of large size it is desirable to pass the gas through a condenser between the two stages of compression in order to remove some of the heat, reduce the volume and save power. This is done with the York compressor, shown in section by Fig. 66, where the connecting pipes are seen at the top, and by Fig. 70, illustrating the machine complete. The large compound compressor made for the author in 1884—of which Fig. 67 is part section of the h. p. cylinder—had a tubular condenser interposed between the two stages of compression, and it gave most excellent results, the diagrams showing nearly isothermal lines. In the low pressure cylinder of Fig. 66 it will be noted that the valve in the piston is annular, and thus it requires only one-half the lift of an ordinary mitre valve to give the same area of discharge. In the Auldjo compressor, Fig. 60, owing to the valve in the piston being fast on the piston rod itself, and the piston being loose on the rod, the amount of its opening will have to be deducted from the nominal stroke, to arrive at the effective length, because the actual stroke of the piston will be so much less than the stroke of its rod.

It would seem at first sight a self-evident proposition that a compressor and its steam engine should be combined as one machine. But as a matter of fact such well known and largely used types of refrigerating plants as the "Linde" and "Hercules" are built up from the two machines made separately (numbers have come to Australia with their engine and compressor built by two entirely different makers)—see Figs. 71 to 73—and in such case they of course require double foundations and extra careful erection. There must be some reason therefore for this separation which should repay our investigation, to do which we must

go back a little, and again consider the work to be done by the piston of a compressor in its relation to the work of a steam engine.

THE WORK TO BE DONE BY A COMPRESSOR PISTON.

Fig. 74 represents a diagram or indicator card as taken from an ammonia compressor by an eminent firm of refrigerating machine builders, who claim to obtain nearly isothermal compression under their system of injecting oil at every stroke of the machine. From this diagram it will be seen that there is no effective work performed by the piston at the commencement of its stroke when the pressure on both

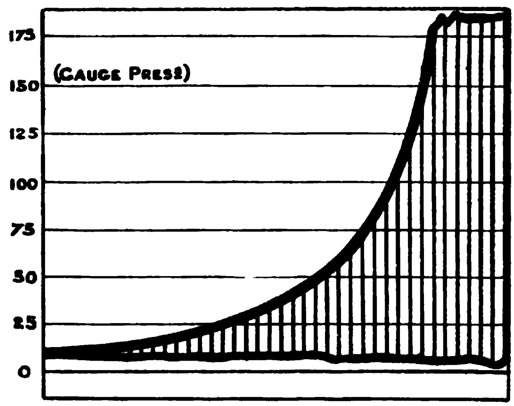


FIG. 74.—INDICATOR CARD FROM AMMONIA COMPRESSOR.

sides is the same; at quarter stroke the pressure against it is equivalent to about ten pounds per square inch, at half stroke about thirty pounds, at three-quarters stroke, 100 pounds, and the maximum pressure, about 180 pounds, is reached at about five-sixths of the stroke, when the delivery valve opens, and the pressure thence continues uniform during expulsion to the end of the piston's journey.

A diagram of the work performed against the piston of an expansive steam engine is, of course, just the reverse of such a compressor diagram, because in the engine the maximum work is at the commencement of the stroke, whence it continues practically uniform until the steam is cut off, and

then it diminishes gradually toward the end, in accordance with the grade of expansion at which the steam is worked.

Air compressors, such as Fig. 75, do not usually work to as high ratios of compression as ammonia machines do, and

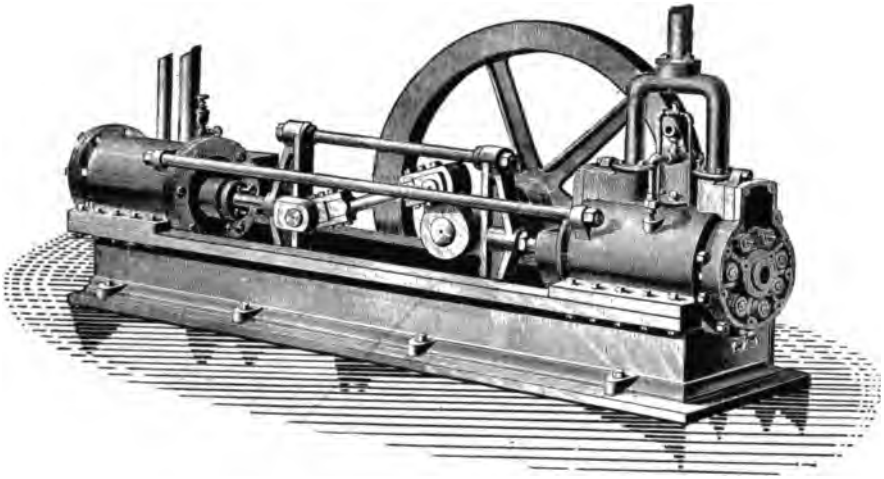


FIG. 75.—STRAIGHT LINE ENGINE AND COMPRESSOR.

most builders of them stick fast to this "straight-line" system. Fig. 76 represents two indicator diagrams taken one each from the engine and compressor cylinders of the same straight-line machine and superimposed. By this it is

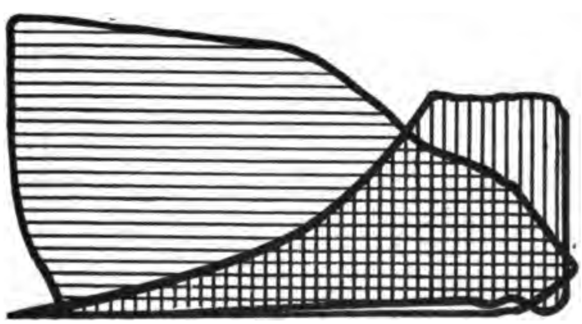


FIG. 76.—INDICATOR CARDS FROM STEAM AND AIR CYLINDERS.

shown clearly how unequal is the relative effort and resistance at different parts of the stroke. The small portion covered by crossed lines represents the whole portion of the work which is transferred direct from the piston of the

engine to that of the compressor, although the engine appears to have a slide valve and carries the steam well past half stroke.

These discrepancies are greatly intensified if we take the compressor card, Fig. 74, and superimpose upon it the card from a Corliss engine, cutting off at from one-fifth to one-quarter stroke, as in Fig. 77.

A B C D E A is a diagram from a Corliss steam cylinder hatched with horizontal lines, and F G H D F is the diagram from the compressor just referred to hatched with vertical lines. That part of the figure which is covered by

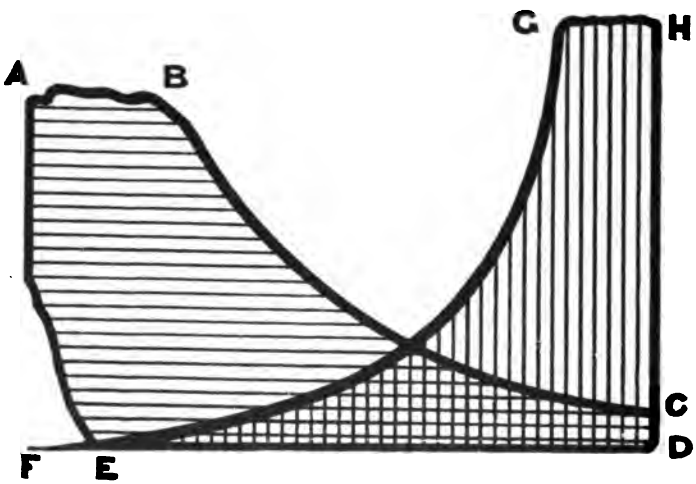


FIG. 77.—INDICATOR CARDS, CORLISS ENGINE AND COMPRESSOR.

the intersected lines represents the very small portion of the whole work that would be communicated directly from the piston of the engine to the piston of the compressor if the two were coupled up in a straight line; that proportion of the work which is shown by plain horizontal lines would have to be delivered into the fly-wheel at the early part of the stroke, and the work represented by the area covered by plain vertical lines would have to be given up again by the fly-wheel to the compressor piston at the latter end of the stroke. All these points have to be considered before we can properly investigate the construction of the whole machine, steam engine and ammonia compressor combined.

GENERAL DESIGN AND CONSTRUCTION OF THE WHOLE MACHINE.

FOURTHLY.—*Other things being equal, the machine should be all self-contained, be easily erected, and require the minimum of foundations.*

FIFTHLY.—*As double-acting compressors require only one-half of the stress on their connecting rods, bearings and cranks, that is necessary with single-acting ones of equal capacity and stroke to do the same amount of work, then all compressors should be double-acting, unless there are insuperable disadvantages connected with such an arrangement. And, as the smaller the ratio of compression, the more equable is the work of the piston, there may be manifest advantages in compressing by stages a few ratios at a time, if not accompanied by increased complication and cost in other directions.*

The writer, from a life's experience of machine builders and machinery users, is inclined to the belief that purchasers often think that they can do without old-fashioned advice, and imagine that they are keen buyers, when they make a saving by paying a few dollars or pounds less for one machine than they are asked for another, which to their ideas is a similar one. Such people often find out afterward that they made a mistake, because they did not sufficiently value some old-timer's experience, or take into consideration what the relative cost *erected complete* and upon their foundations, ready for work, of the different machines offered to them would come to, and understand that annual up-keep and wear and tear are important factors.

SIXTHLY.—*The engine piston should be connected directly to the piston of the compressor, and the cranks, connecting rods and bearings of the machine should only transmit the* DIFFERENCE *between the engine force and the compressor resistance instead of the* SUM *of the work represented by the two.*

This would appear as a self-evident proposition to be universally followed, as it is done in straight-line compressors, were it not for the teaching of preceding paragraphs, which show how the great want of correspondence between the power of the engine and the resistance of the compressor, during the cycle or revolution of the crank shaft, necessitates enormous fly-wheels, and increases the frictional losses.

SEVENTHLY.—*The pistons and valves should be easily accessible for examination and removal.*

Horizontal valves are generally easily accessible, but they want looking to so much oftener than vertical ones, that the makers of the machine shown by Fig. 63 go to great expense to fit vertical valves into cages, which are again fitted into horizontal recesses in the cylinder; but in the same machine there must always be a little bit of a picnic if one of the piston valves gets stuck. In Fig. 61 the difficulty is only apparent and not real, as three valves can be withdrawn by removing the top cover, and the low pressure inlet valve is accessible from the bottom door. The low pressure delivery valve is made so as to withdraw right through the trunk and high pressure piston.

A thoughtful inspection of some of the several compressor cylinders illustrated reveals interesting features, which suggest a number of questions—for instance:—Why do makers of some compressors put their outlet pipes on to the covers or heads of their machines in such a way that neither the piston nor the valves can be got at without breaking a great number of joints and taking down what should be permanent connections?

The writer was once shipmate with a steam crane built by makers who were very eminent for certain classes of machinery, but were apparently starting a new line of "grab" cranes; well, this crane had the steam pipes screwed into the doors or bonnets of the slide valve chests, but, owing to the absence of flanges, it was necessary to take off *eleven* separate pieces of pipe to get to those slides. If compressors of this class are not expected to call forth expressions, at times, which are more forcible than poetic, they will have to be placed in charge of very *good* men in more senses than one.

EIGHTHLY.—*Covers or bonnets should be made with simple joints and no bridges.*

This, like the seventh condition, will be best illustrated, perhaps, by instances in which the condition is not fulfilled. Fig. 57 is a section of a compressor which has a great number of extremely good points as a machine, but it also has

triple face joints under the heads on the lines A B. This arrangement necessitates most accurate workmanship in the fitting, and extreme care when making the joints at the two bridges, to insure that they do not blow through. A leak in such a case may be going on for a long time before it is found out. The author's two compressors, Figs. 68 and 69, are sinners on this point, but as he is now twenty years older than he was when he committed the offense, he has lived long enough since to see the error of his ways.

Figs. 59 and 65 show similar joints on their compressor heads; these, like the pipes direct on to the heads, are not necessary at all, unless it is desired that the man who has to make the joint in a hurry and be responsible for it afterward should become an adept in profane language. A joint is of course a relatively simple matter to make, now that flanges are faced by high-class tools, to what it was formerly. The author has made joints of curious material in his time, such as, for instance, tinfoil and blotting paper on ether machines, well kneaded dough from wheat flour for cold kerosene, fire-clay and red lead for hot oil, and all such nostrums, which were the best things known for their respective purposes at one time, before the present great army of patent packing people made life easier. With all these to hand, he has found nothing better for a compressor head or any other ammonia joints, than a thin lead gasket placed in a recess where it cannot get away. To make a sure success, all bonnets and flanges should be plain circular, and turned for the rings. If people will make simple flat surfaces and use jointing material which will squeeze out and get in the way of their valves or pistons, they must expect trouble sometimes; but such old-time rough-and-ready methods are not good practice now. The jointing material, whether metallic, fiber, rubber or insertion, should be inclosed where it cannot spread. For examples of joints see the covers in Figs. 58 and 62, which show two separate ways of keeping the jointing from spreading.

ENORMOUS FLY-WHEELS.

It is evident from the foregoing illustrations that builders of compressors have good reasons for the employment of

the extremely heavy fly-wheels which often distinguish this class of machinery. Such wheels require heavy shafts and journals, and therefore greatly increase the friction in the bearings and the power necessary to drive a given sized machine, and also add to the first cost and maintenance when at work. This being well understood, there has in consequence been plenty of inventive skill displayed in devising compressing machinery with all sorts of arrangements to enable the work performed by the steam piston to coincide more nearly with the work required by the compressor's piston at every part of the shaft's revolution.

There is a great deal of popular misconception with regard to the power wasted in driving fly-wheels, it being often stated that such power is only required at first starting them into motion. The actual horse power continuously expended is represented by the formula:

$$\text{Horse power:} \quad H = \frac{f\,W\,S \times .26d}{33{,}000}$$

Where f represents the co-efficient of friction from .03 to .25 in wrought iron upon gun metal lubricated, it cannot safely be taken at less than .05 in actual continuous work; W the weight of fly-wheel, S the speed or revolutions per minute, and $.26d$ the circumference of the journal in feet when $d=$ the diameter in inches.

Take for example a 5-ton fly-wheel making 90 revolutions per minute with 9-inch journals, then $9'' \times .26 = 2.34$ ft. cir. of journal $\times$ 90 revs. $=$ 210.6 feet per minute. Five long tons $=$ 11,200 lbs., which multiplied by .05, $=560$.

$$\text{Then } \frac{560 \times 210}{33{,}000} = 3.5 \text{ horse power.}$$

With fuel evaporating 8 lbs. of water per minute and an engine using 30 lbs. of steam per horse power per hour—

$3.5 \times 30 = 105$, and $\dfrac{105 \times 24}{8} = 315$ lbs. of coal wasted every twenty-four hours simply to drive the wheel.

RIGHT-ANGLED ARRANGEMENT OF ENGINE AND COMPRESSOR.

In order that the continuously varying power of the engine during the course of a stroke or revolution, may be

applied in such a way as to correspond better with the work to be done, and be more effective at the time when the compressor piston offers the greatest resistance to it, great numbers of refrigerating machines are now built in such a way that the effective axis of the steam cylinder with regard to the crank shaft is at right angles to the axis and stroke of

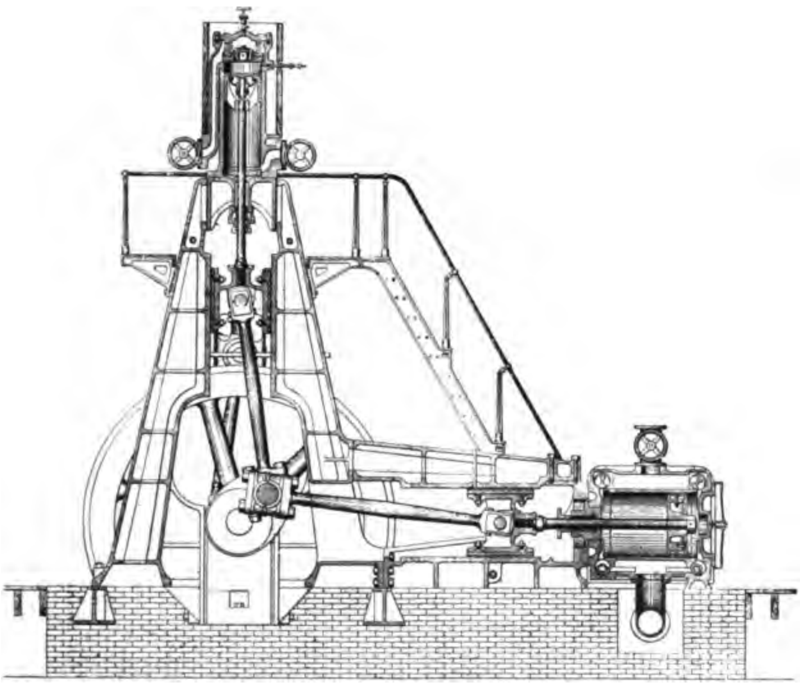


FIG. 78.--SECTION OF FRICK CO.'S ENGINE AND COMPRESSOR.

the compressor, and this is generally carried out under one or the other of the following arrangements:

Under the first one, the two connecting rods from the crossheads of the engine and compressor respectively, are connected to the same crank pin, and thus transmit the power without any torsion on the shaft, as seen in Figs. 78 (Eclipse) and 79 (De La Vergne), which represent American machines of the very highest class, having horizontal engines and vertical compression cylinders. Examples of the other arrangement are shown by the Australian compressors, Figs. 4 and

54, where the steam engine is vertical and the compressor cylinder horizontal.

Under the second plan the compressor is set parallel to its engine, which is often on an entirely independent foundation, especially when the two machines are both horizontal. Two separate cranks are provided, one for the engine and the other for the compressor, which are keyed on to the opposite ends of the fly-wheel shaft at an angle of 90° or thereabouts. See Fig. 71 for a typical example which should

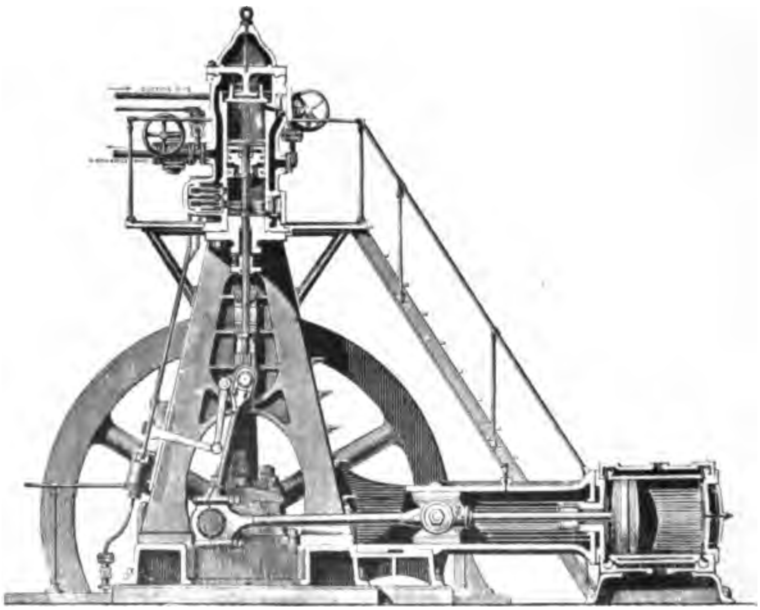


FIG. 79.—SECTION OF DE LA VERGNE ENGINE AND COMPRESSOR.

be carefully compared with Fig. 54, because the compressor cylinders are practically the same in the two cases. The operation of the engine on the compressor is nearly the same in both these machines; but necessarily there is in Fig. 71, besides the torsion on the shaft, more main bearing friction, additional first cost, and double foundations to be provided.

An examination of the double-acting compressors, Figs. 54 and 71, will show that in both cases two single-acting horizontal cylinders could be substituted for the double-acting one, without in any way affecting the relation of the steam

engine piston to the motion and effective power of the compressor pistons.

HORIZONTAL ENGINE AND TWO VERTICAL COMPRESSORS.

Two vertical single-acting compressors operated by a horizontal engine require at least two cranks, generally set opposite to one another or at an angle of 180°, in which case one compressor only is driven by torsion of the shaft. Three cranks, however, are often adopted, and entail a great deal of additional complication and expense, which of course the designers of the machines consider justified by compen-

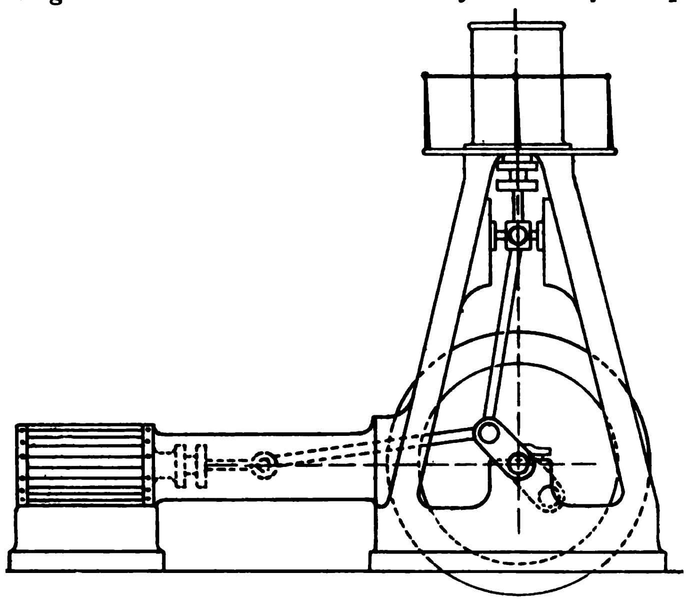


FIG. 80—HORIZONTAL ENGINE AND VERTICAL COMPRESSORS—ELEVATION.

sating advantages, and at least five different arrangements of this type are in common use, all of which have their respective advocates.

Fig. 80 represents the end elevation of such a machine, five different plans of which follow, some having inside and others outside fly-wheels. The advantage of a large fly-wheel is obvious, because if 5,000 pounds weight of wheel can be made as effective as one of 10,000 pounds in a smaller compass, it will only require, as has already been shown, one-half the loss of power to keep it in motion.

To have large inside fly-wheels means very large and heavy sole plates, and therefore some machine builders overhang their wheels at the opposite end to the engine, as in Fig. 81, occasionally extending the shaft for a fourth and outer bearing.

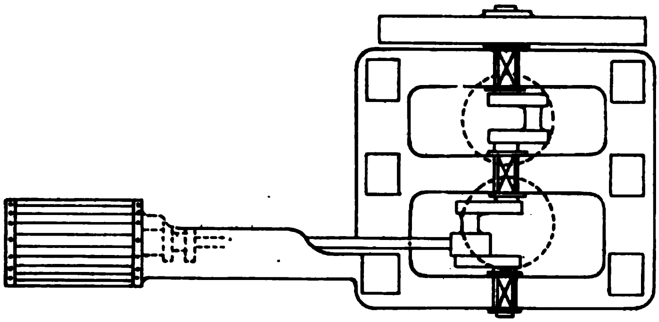


FIG. 81.--HORIZONTAL ENGINE, VERTICAL COMPRESSOR—PLAN.

In Fig. 81 is seen the plan adopted by a very eminent firm of builders. One crank pin, it will be noticed, is of double length, to take the big ends of the two connecting rods. The work in such machines is necessarily very severe on the middle bearing, and, although the shafts are

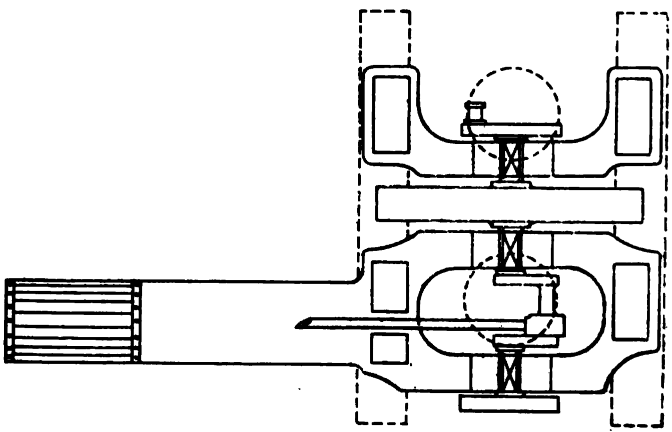


FIG. 82.--PLAN OF FIG. 80 WITH INSIDE FLY-WHEEL.

made enormously strong as compared with steam engine practice, they occasionally fail as a result of the special strains to which compressors are liable.

Fig. 82 shows the arrangement adopted by another firm of world-wide reputation, who put a large fly-wheel between the two compressors and carry the separate portions of the sole plate on massive girders below the floor line. The solid crank, as before, carries two connecting rods, but the outer compressor has a disc crank overhung. If the girders, shown in dotted lines, and the bottom of the separate sole plates are accurately planed, as is no doubt the case, this arrangement is a much better one from a practical mechanic's point of view than the one preceding it.

Fig. 83 shows one single solid crank for the engine and two disc cranks for the two compressors and needs a very large sole plate, as the fly-wheels are inside. With large discs the weight of each compressor piston and its connect-

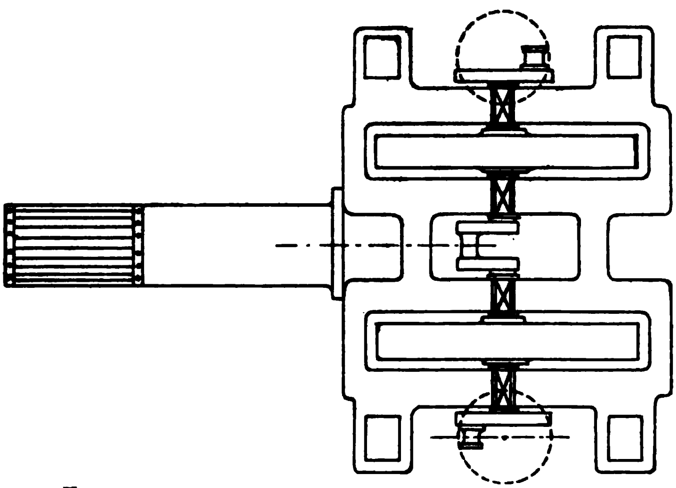


FIG. 83.—MACHINE WITH TWO INSIDE FLY-WHEELS.

ing rod can be balanced separately, instead of in the fly-wheel, and much steadier and smoother running can be assured. The fault of this arrangement is that it requires four bearings to be kept accurately in line; as the bushes wear, the unequal wear which is nearly certain to take place, tends to throw strains upon the shaft and break the crank.

For Fig. 84, the only thing to be said in its favor is that large fly-wheels can be used with a small sole plate; the downward wear of the two outer bearings, however, offers a premium for breakage of the expensive triple crank shaft.

In marine engine practice it is now customary to "build up" these crank shafts, and they are frequently made in short sections with flanges. With triple or quadruple expansion a long series of solid or double cranks and a line of

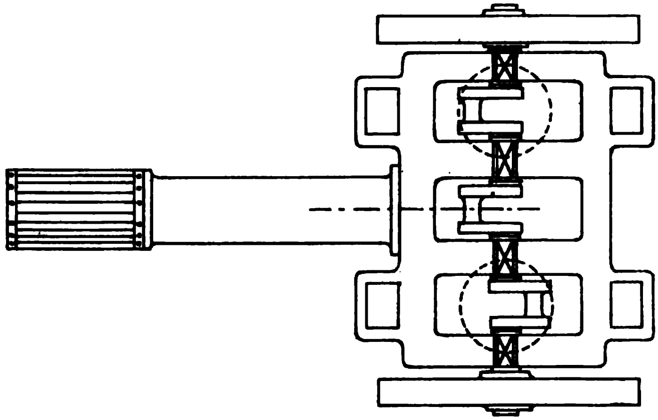


FIG. 84.—MACHINE WITH THREE CRANKS AND OUTSIDE FLY-WHEELS.

bearings are absolutely necessary; but there is no necessity whatever for such complication with a refrigerating compressor. Every experienced engineer knows the advantage of having only two bearings to a shaft, and of that shaft being

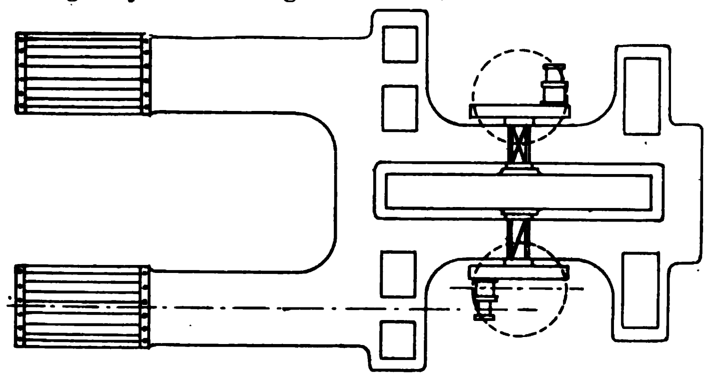


FIG. 85.—MACHINE WITH TWO BEARINGS AND CRANKS, ONE FLY-WHEEL.

a plain one without solid cranks that require the crank pins to be the same size as the shaft itself.

Fig. 85 shows an arrangement of this kind with two engines, preferably cross-over compound cylinders; there is only one fly-wheel between two bearings, and those bearings

should be sufficiently wide apart to prevent the pressure and friction upon them being materially increased as the effect of the leverage due to the overhang of the crank pin. The adoption of a larger diameter of the crank pins for the compressors is optional, but it is mechanically correct. With such an arrangement the steam engine can be made of longer stroke than the compressor by having the two pins eccentric to one another. If space can be afforded in the machine house to give a decently wide spread, there can be no question as to the simplicity and efficiency of this plan of machine.

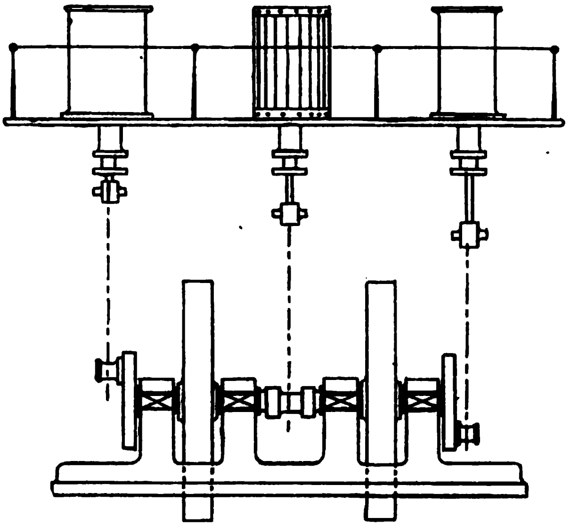


FIG. 86.—VERTICAL ENGINE AND TWO VERTICAL COMPRESSORS.

With vertical or "inverted" engines and two vertical compressors, the adoption of three cranks is imperatively necessary to secure the right-angled action of the engine.

Fig. 86 illustrates one of the most common designs, with two disc cranks for the compressors, four bearings, and two fly-wheels; it would make a better job of it, perhaps, if the outer bearings were larger, the shaft strengthened, and the two inner bearings dispensed with. This type of machine may be modified by making three solid forged cranks with outside fly-wheels put on in halves, or still further varied by putting overhung fly-wheels, making the plan almost a counterpart of Fig. 84, and shown by Fig. 89.

This vertical pattern seems to have been first favored by the great American father of ice making machinery, the late David Boyle, one of whose machines is shown by Fig. 87.

As a contrast to this work of only twenty years ago, and to illustrate by comparison the great advance made since

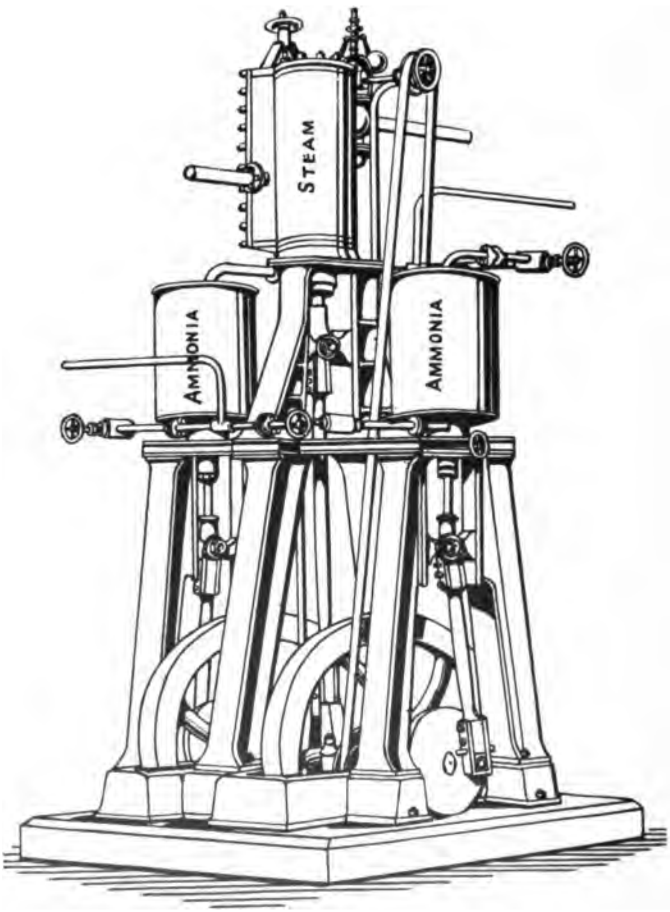


FIG. 87.—AMMONIA COMPRESSOR—ORIGINAL BOYLE PATTERN.

that time, the magnificent machine built by his successors is shown by Fig. 88.

Fig. 89 shows the arrangement with a vertical engine, modified by overhanging the fly-wheels.

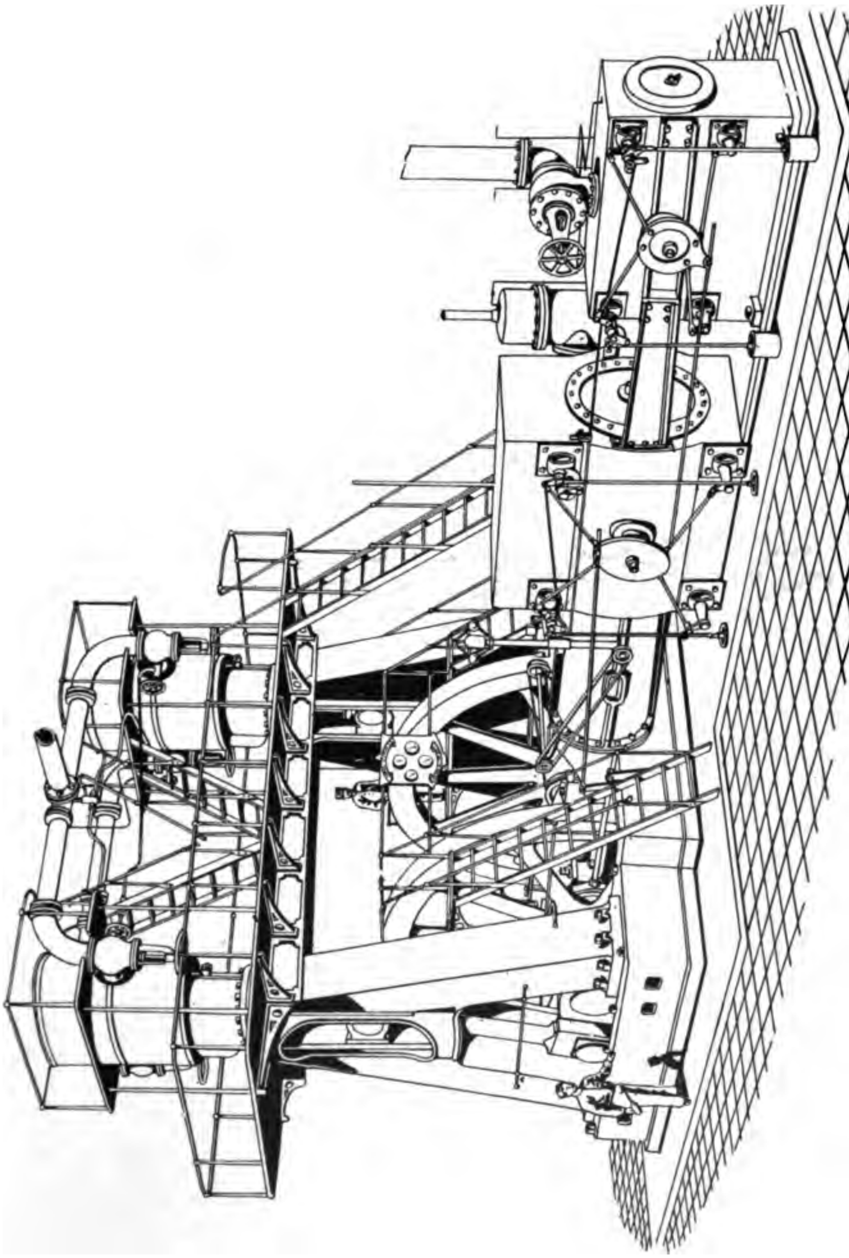


FIG. 88.—LARGE SIZE MODERN BOYLE MACHINE WITH COMPOUND TANDEM ENGINE—PENNSYLVANIA IRON WORKS CO., PHILADELPHIA, PA., U. S. A.

In all the accompanying illustrations where the effective axis of the engine is at right angles with that of the compressor, the engine is on its dead centers (and therefore exerting no power directly from its piston) at the time when the compressor piston is just below half stroke; so that the motive power in such positions must come from the fly-wheel. A little examination will also show, that as the crank comes toward either the top or bottom centers, and, with the compressor connecting rod, approaches the vertical position, then, the centers of the crosshead pin, the crank pin, and the shaft are coming into line together, which constitutes a toggle joint of the crank and connecting rod. The action of

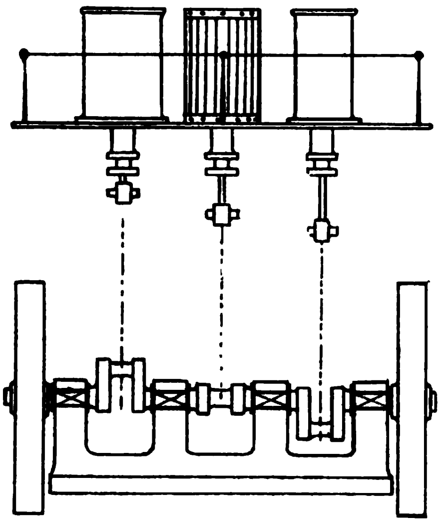


FIG. 89.—VERTICAL MACHINE—OVERHUNG FLY-WHEELS.

the engine on the central pin of this toggle is to create a gradually increasing force, which approaches the theoretically infinite at the two compressor centers, just when the compressor pistons offer the greatest resistance.

DIAGRAMS ILLUSTRATING RIGHT-ANGLED CONNECTION.

The actual effect which is produced on the distribution of power from the piston of the engine to that of the compressor, whether arranged in one or other of the ways shown by the several machines illustrated, is graphically and effectively shown by Fig. 90, a diagram from a Corliss engine and

ammonia compressor, which is merely a transposition of what is seen on Fig. 77.

In this diagram, Fig. 90, the length of the base line represents the travel of the piston, or the stroke of the machine; and the vertical heights from any points on the base to the curved lines, the relative pressure on the pistons in such positions. The compressor diagram—hatched with vertical lines— is identical with that on Fig. 77, but the transposition of the varying pressures shown by the steam engine card is

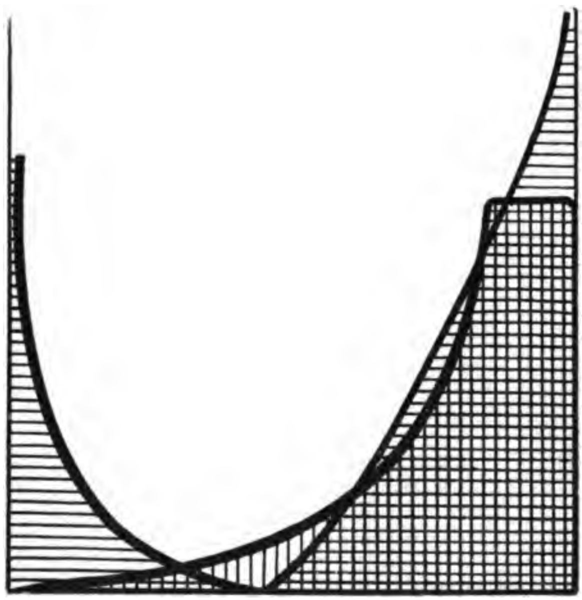


FIG. 90.—DIAGRAMS, CORLISS ENGINE AND COMPRESSOR (TRANSPOSED).

so radical, that it would not be recognized without explanation. The portion hatched by horizontal lines, however, represents the equivalent in energy of the engine power, as on Fig. 77, but so transferred as to correspond with the motion of the compressor crosshead instead of its own.

The horizontal base line from left to right represents the stroke of the compressor piston from the bottom to the top center. The power of the engine on the compressor connecting rod and piston is at its maximum at the commencement of the stroke, as the engine is then a little past half stroke;

but this power comes down to nothing at about half compressor stroke, when the engine arrives on either of its own centers. It will be noted that this center point of the engine is not exactly at midstroke of the compressor, but is nearer to the left side; this is owing to the angle of the compressor's connecting rod shortening the height of its crosshead. At the right hand side of the figure the engine is again out a little past half stroke, due, as before, to the angle of its own connecting rod, and the compressor is then on its top center. In this figure, it will be seen, nearly all the compressor's diagram is overlapped, and covered by the crossed lines, and

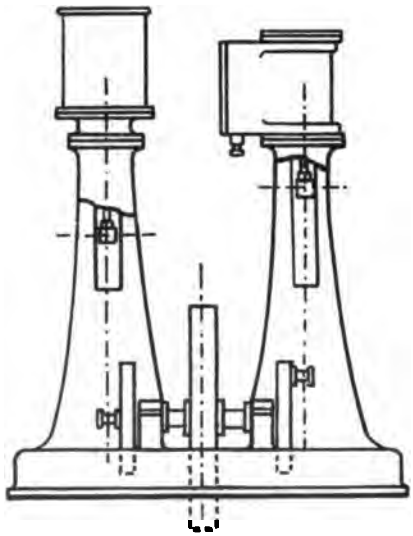


FIG. 91.—ENGINE AND SINGLE-ACTING COMPRESSOR.

the beautiful effect of the right-angled connection is made very clear. Sufficient bare horizontally hatched space is left to represent the surplus power which is required to cover the frictional losses, and it is evident that, other things being equal, any arrangement in which the work to be given and taken as is shown in Fig. 90, will only require a small fraction of the fly-wheel power storage which would be necessary in a case such as is indicated in Fig. 77. The wide adoption of a machine in which a horizontal Corliss engine is combined with vertical compressors is thus seen to be fully warranted by theory as well as by the result of practical work.

In the case of small machines made for dairies, and for butchers' use, as in Fig. 91, there is often only one compressor, and that single-acting, combined with a slide valve engine. In such case one-half of the work of the engine at least must be put into the fly-wheel.

FIG. 92.—DIAGRAM FROM MACHINE LIKE FIG. 91.

Fig. 92 shows the application of the power of the engine to such a compressor; the power as before being hatched with horizontal lines, shows the engine cutting off at three-quarters stroke. The compressor work is covered by vertical lines.

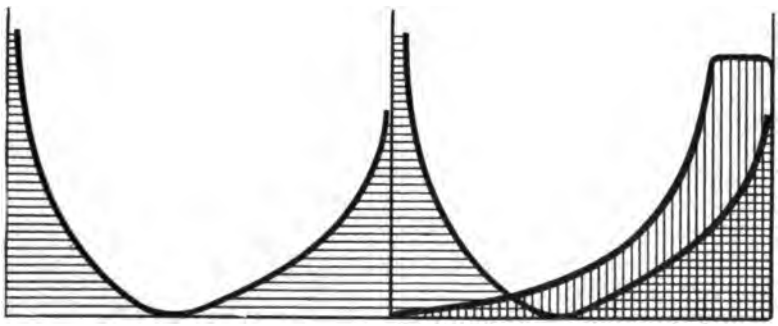


FIG. 93.—DIAGRAMS FROM FIG. 91, WITH RIGHT-ANGLED CRANKS.

Fig. 93 gives the diagrams of the same machine's work, but with the cranks at right angles.

HOW TO PLOT DIAGRAMS OF A COMPRESSOR'S WORK.

As it may not be clear to every reader how the preceding diagrams have been constructed, and as the graphic method adopted may be used for other purposes, such as for

ascertaining the loss by friction in a complex machine, and as such a method of investigation will settle scientifically many questions, the answers to which are often only guessed at, the large diagram, Fig. 94, is introduced to illustrate the work of a Corliss engine and pair of compressors.

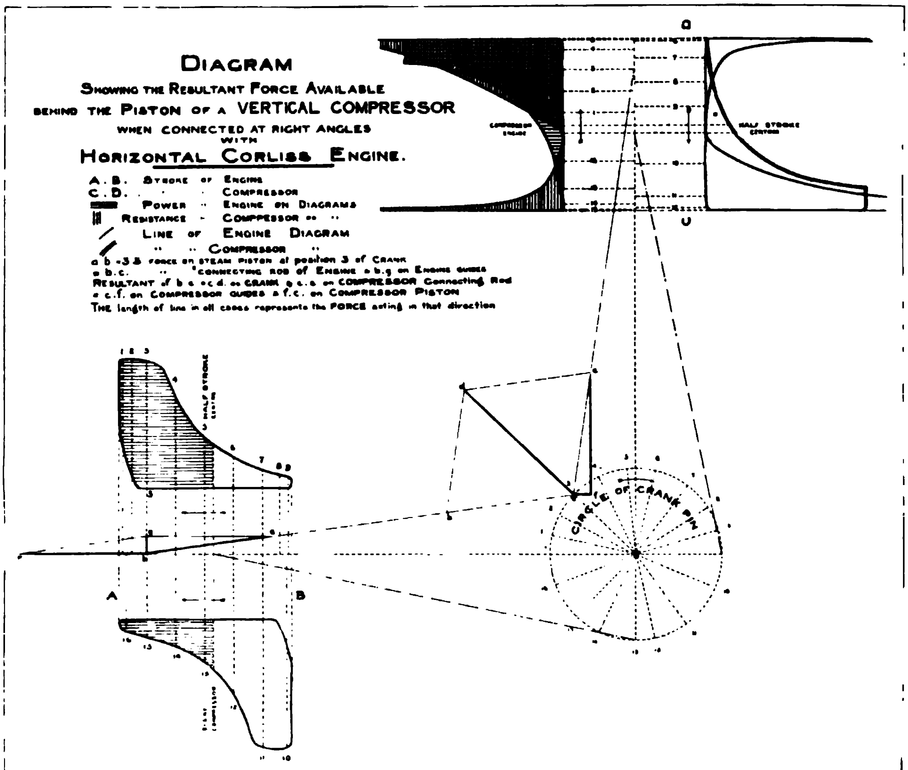


FIG. 94.—DIAGRAM ILLUSTRATING WORK OF CORLISS ENGINE AND TWO COMPRESSORS.

In this diagram sixteen positions are taken in the path of the crank pin, besides the engine centers, and the top center of the compressor. To save space the connecting rods are centered direct from the crank pin on to the piston centers. The length A B represents the stroke of the engine and C D the stroke of the compressor. To prevent confusion which would result from showing the maze of lines necessary to work out the whole of the nineteen positions, only those

are given which have reference to one position (No. 3), although the same work has been done for the whole nineteen positions of the pistons.

The arrow on the crank pin circle shows that the engine runs "overhand." The compression diagram on the left side which is hatched belongs to the compressor working off the engine crank, and which compresses while such crank is passing from position 13 to the top center. The compressor diagram, in double line, on the right side, belongs to the other and opposite crank, and the compression there takes place while the engine crank pin is passing from the upper to the lower center. When the compressors are on their two centers—one at the top and the other at the bottom —the engine crosshead, owing to the angle of its connecting rod, is not at half stroke, but is considerably nearer to the crank shaft. This angle of the connecting rod causes a good deal of inequality in the work directly available for the two compressors, and makes work for the fly-wheel if the cut-off is the same at the two ends of the engine. The compressor on the engine crank is clearly seen to get more engine power than its fellow, because the portions of the two engine cards which are covered with horizontal lines are much larger in area than the plain portions belonging to the other compressor. The space between the lines marked "engine half stroke," and "compressor centers," is added to one and taken from the other, by the inclination of the connecting rod.

The position taken for full illustration is No. 3, where the crank is at about an angle of 45°, and the engine piston is under full pressure, having completed one-sixth of its outstroke. The height of the upper diagram at 3.3 in the outstroke or back-end card, represents the pressure on the piston, the area of which is assumed to be unity. This measurement representing the force acting against the piston is transferred to a b, on the line of the piston rod, and by the construction of the parallelogram a b c g gives b c as the thrust on the engine connecting rod, and b g as the pressure on the guides. (By taking the pressure on the guides in all positions an estimate can be made of the frictional losses due to the varying angle of the connecting rods.) The

length b c at the crosshead end of the connecting rod is transferred to b c at the crank end, c being the center of the crank pin. By the construction of the parallelogram b c e d with d e in line with the crank centers, then the length of c e represents the amount of force or thrust on the compressor connecting rod, and c d the direct downward angular thrust on the main bearing; the latter being the resultant of the separate stresses on the two parts of the crank pin. By drawing e f vertically from the point e, with c f horizontal, the length of the former line, e f gives the amount of the direct vertical force of the engine available for the work of the compressor, and c f represents the pressure of the crosshead against the compressor guides. At the position 3 on the compressor diagram, where the top end of the connecting rod is centered, a line equal in length to f e is set up as representative of the pressure or force available to move the compressor piston in that position. By drawing a similar series of parallelograms to every one of the other positions the corresponding lengths of line have been found which enable the complete diagrams to be constructed. No allowance or deduction has been made in any of these cases for friction; but it is evident that if the co-efficient of friction is known, then the actual loss, and the mechanical efficiency of the whole machine, can easily be ascertained by the same method of investigation.

DIAGONAL CONNECTION.

It has been shown that a steam engine with a very early cut-off is specially applicable for a right-angled connection with a compressor; but a comparison of Fig. 90 with Fig. 93 makes it clear that a later cut-off is not so well fitted for the purpose, because there is a much greater proportion of power than required at the beginning of the compressor's stroke. This can be rectified by making the connection diagonally at some other angle than 90 degrees.

If, as is no doubt the case, there are still many refrigerating engineers who question the advantages that are claimed for compound compression, there are but few, and certainly none among those who have lengthened experience with compound engines, who fail to duly appreciate the

effects of compound expansion. Where the load is steady, as in a refrigerating machine, a tandem compound has many advantages over a single cylinder Corliss engine; it gives more even running with smaller fly-wheel, and requires less working expenses to make good the wear and tear.

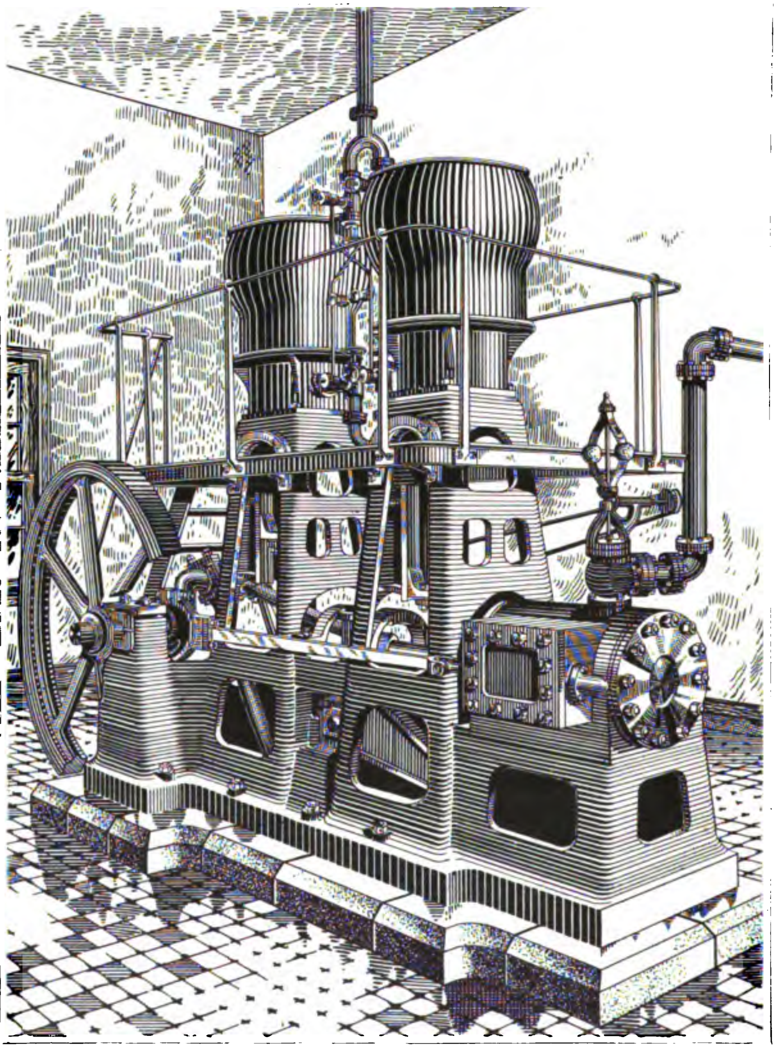
It is well known that no form of engine has less loss by friction than a beam engine; and when the connecting rod big-end moves in the arc of a circle, with a versed sine of only an inch or two, instead of in the circle of the crank pin path, then friction on crosshead guides is reduced to a minimum.

Having been the first engineer to introduce and design tandem compounds in Australia, the author may (without knowing it) be a little prejudiced in his preference for them; be this as it may, he thought some short time since that it might be possible to arrange a pair of single-acting compressors with a single slide valve engine—either simple or compound—under a new design which should by the adoption of levers unite all the best features of a modern machine in a simple and effective combination, in which the engine and the two compressors should be all in line with one another, and erected on one compact sole plate and foundation.

The machine as designed, for better or worse, is shown in perspective on following page, and in sectional elevation by Fig. 95, and is now open to the free comments of machine builders and machine users, whose criticisms, however harsh, will be gladly welcomed if genuine. No machine is perfect, and this one has many points to which exception will be taken; still, it is only by the gradual elimination of faults that any machine approaches that perfection to which it can never arrive.

An inspection of the two figures will show that there is a single horizontal engine—by preference for large machines a tandem compound—which is made with a high foundation plate, so as to afford space in which to carry a lever, rocking beam, or bell crank, centered right under the guides. There is only one single bent crank on the shaft, but with an extra long crank pin, this crank shaft may have a fly-wheel on one or both sides of the machine, but is never subjected to tor-

sion other than that due to the work of the fly-wheel, which, as will be seen later on, is extremely small. The interven-



PERSPECTIVE VIEW—ANTARCTIC REFRIGERATING MACHINE—BEAM PATTERN—TEN TON.

tion of the rocking levers at different angles to the main center, and the different angles of the two connecting rods

with regard to the crank pin, puts the steam piston so far behind that of the compressor, that when the engine is at half stroke the compressor piston has completed six-sevenths of its journey. The combination of the connecting rods and cranks forms a most effective toggle joint, and they operate on both compressors without any torsion on the shaft. As the strains are all on the one center line, and the

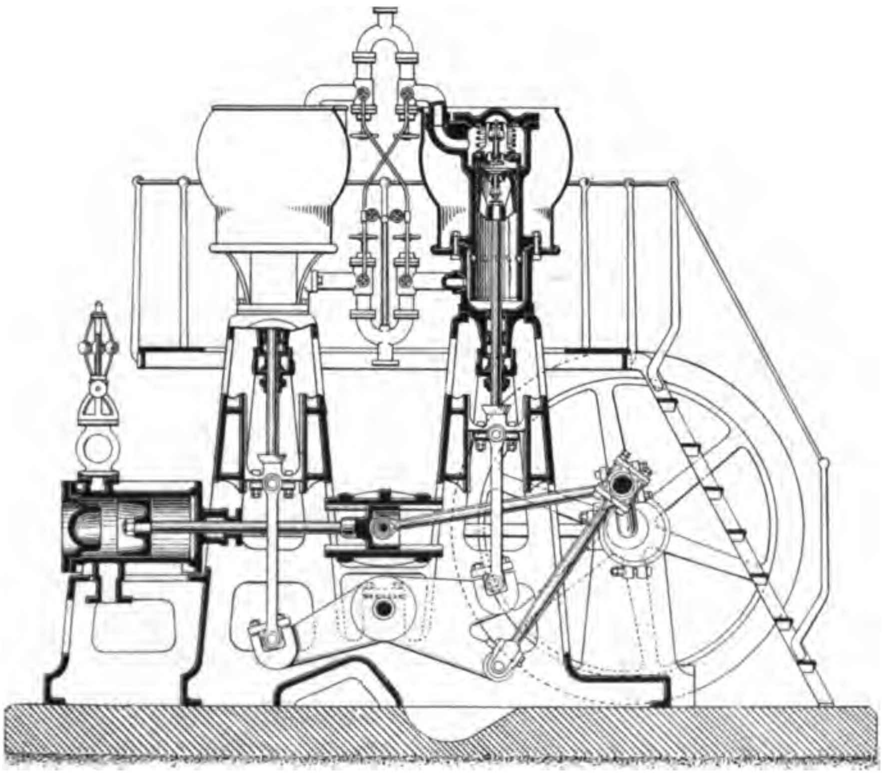


FIG. 95.—SECTION OF BEAM PATTERN ANTARCTIC COMPRESSOR.

machine is self-contained, hardly any foundation is necessary. As the connecting links to the compressor crossheads hardly move an inch out of the compressor's vertical line, the friction of the guides is only nominal. The adjustment of clearance is easily effected by lining under the bushes at the ends of the main lever, thus dispensing with the nuisance of screws and nuts. The piston rod and connect-

ing rod of the engine work between the compressor links. The compressor cylinders themselves are shown on a larger scale in Fig. 58, and it will be noted that they are plain barrel castings, the belts or chambers around the bottom ends being cast separate with the stuffing-boxes.

The resolution of the forces in this machine has been worked out on the same principle as those shown by Fig. 94, and the diagram produced is given by Fig. 96. In this the same compressor diagram is used as before, but it will be

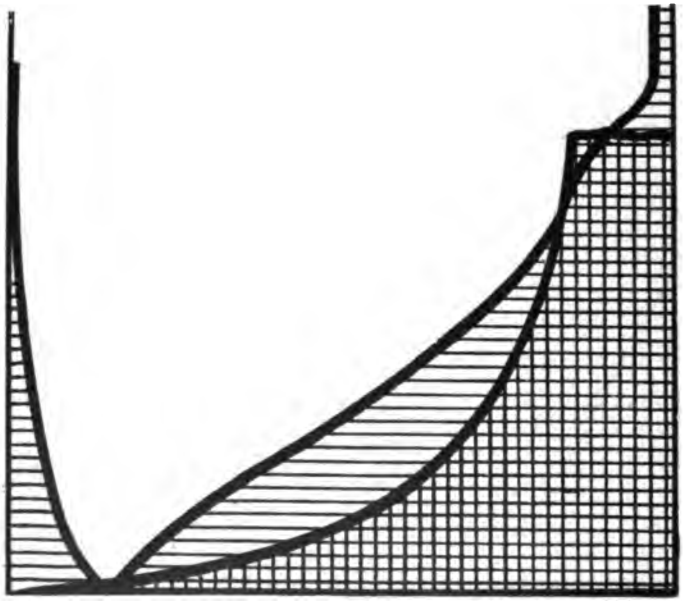


FIG. 96.—DIAGRAMS OF ENGINE AND COMPRESSOR FROM BEAM MACHINE.

noted that the point where the engine is on the center, and where there is no power to be given off to the compressor, is much nearer to the commencement of the compressor's stroke than in the right-angled arrangement. The engine power in this diagram is drawn a little excessive, perhaps, for comparison with the work of the compressor, but it will be noted that the compressor card is all but entirely covered by the horizontal lines.

This design can be modified by putting a high and low pressure steam cylinder side-by-side instead of in tandem,

and for very large machines the levers would obviously be well below the floor line.

GEARED COMPRESSORS.

Some English builders of refrigerating machines favor the use of gears, and build double-acting horizontal compressors driven by means of spur gear from a horizontal engine running at a higher speed. The saving by this arrangement, if any, arises from being able to use a small steam engine, running at a greater number of revolutions per minute than the compressor. Against this has to be set all

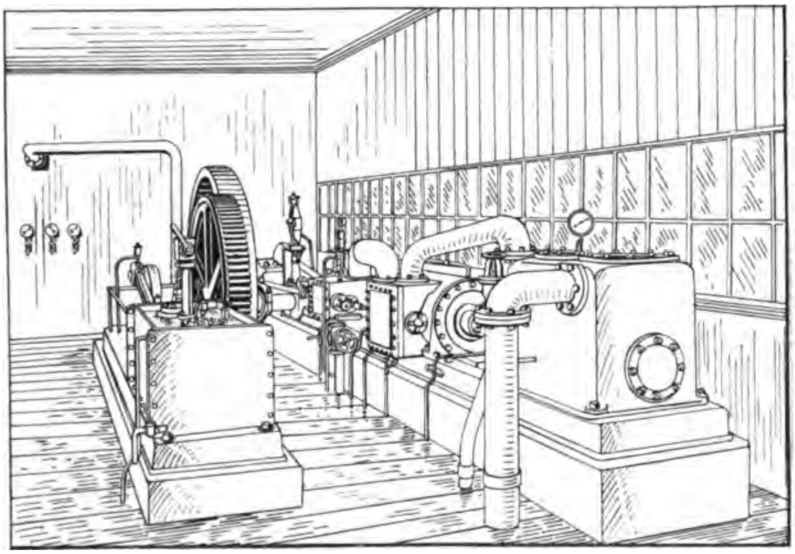


FIG. 97.—HORIZONTAL COMPOUND CONDENSING ENGINE GEARED TO HORIZONTAL COMPRESSOR.

the complication of extra shafting, and the noise and friction of the gearing. It is extremely doubtful whether this form of compressor can show a lower consumption of steam for the same weight of ammonia compressed than the best directly driven machines. Such an arrangement must take up an immense floor space, as seen by Fig. 97, and for obvious reasons it is not likely to have many imitators, the more so, as later developments in machine design permit of a much higher piston speed for compressors than was pos-

sible in old forms, which are restricted in their delivery through having both the inlet and outlet valves in the top covers or heads of the cylinders.

In other arrangements horizontal compressors are geared to a vertical engine, and vertical compressors to horizontal engines, but they appear to be principally confined to English practice. In a large London brewery there are three pairs of compressors set all in a line, each pair having a mortise-wheel gearing into an iron pinion on the main driving, or extended engine, shaft. As the compressors are of the old fashioned type with two valves in the head, the maximum speed is fifty-five revolutions, and the gearing is as 2 to 1. As the stroke is only fifteen inches there is no doubt that with more modern valve arrangements these compressors could be driven direct from the engine. It must not be forgotten, however, that there is an advantage in being able to put one, two or three pairs to work as the demand for cold arises, and that the risk from break-down of a compressor is minimized if your engine is never to be laid up. With machines from experienced builders there does not seem to be any reason why the compressors should not be as reliable as the engine.

BELTED COMPRESSORS.

No account of refrigerating machines would be complete without a chapter on belted compressors, for while separately they may be of comparative insignificance when compared with the giant steam machines, running up to as high as 500 tons capacity, they are in the aggregate of immense importance, owing to their more widely extended use. The development of modern creameries and dairies, with their steam driven separators and churns, has necessitated in the majority of instances, the addition of a refrigerating machine of proportionate power. In the case of advanced retail butchers who employ steam choppers and other machines, and who, like the dairy men, need refrigerators to keep pace with the times, the line shafting generally fitted up on the premises enables a small refrigerator to be simply driven by means of pulleys and belts. To meet the demand which has thus sprung up, there has been a great increase in the

designs for small plants, and their makers now may be reckoned by hundreds.

Belt driven compressors may be broadly classified under two divisions, namely, the "open" and "inclosed." About the former class very little need be said, as any of the types of compressing cylinders already referred to may be fitted up with pulleys on their crank shafts, instead of having a steam engine directly coupled to the same crank or a separate one. Such machines need to differ in no other way from an ordinary steam driven compressor, but it is obvious that with only one single-acting compressing cylinder, a very heavy fly-wheel is necessary, because in such case the work is all concentrated in about the sixth part of a revolution. The work of the piston shown by the indicator card from the compressor, when transformed by the action of the connecting rod and crank, and bent round in the circle of the crank pin, would appear somewhat as Fig. 98, where the radial lines in the lower diagram correspond to the work of the indicator diagram above, a rectangle equal to the distance between the two outer circles, D E, multiplied by the length of the inner or crank pin circle, representing the mean work of the belt. The uncrossed circular lines cover the area representing the work which has to be put into the fly-wheel, while the uncrossed radial lines show the work that must be taken from the fly-wheel, if the work of the belt is uniform. The mean length of these radial lines multiplied by the length of the arc of the circle they stand upon (or the area of the cam shaped figure, if the circle was opened out to a straight line) is exactly equal to the area of the compressor diagram. The diameter of the crank pin path, C D, is exactly the stroke of the compressor, A B.

As the power transmission capacity of a belt is uniform it is evident from the above illustration that in the absence of a fly-wheel, such a machine would require belt power to be provided about six times as great as would be necessary if there were uniform resistance. With two single-acting compressors combined, or with one that is double-acting, the working is more regular, as there are two cycles in a revolution, and appears as in Fig. 99, to which the explanation of Fig. 98 applies, it being noted that the circular lines now

cover twice as wide a space as before, representing double the belt power, that is, the rectangle formed by the length of the crank pin circle multiplied by the distance D E. The lengths of the arcs on which the radial lines stand, multiplied

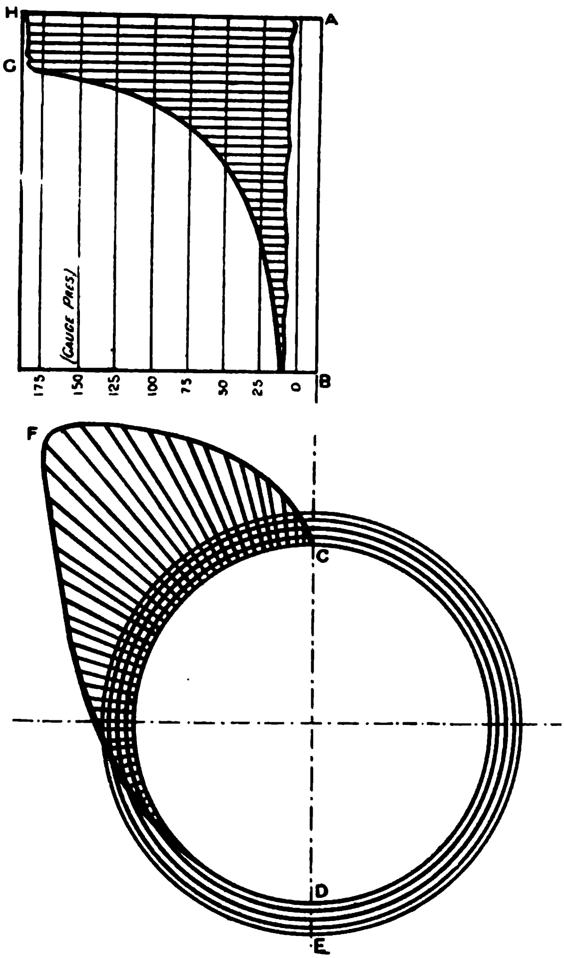


FIG. 98.—DIAGRAM SINGLE-ACTING BELTED COMPRESSOR.

by the mean lengths of such lines, represent, as before, areas exactly equivalent to the area of the two compressor cards above, and the maximum resistance of the compressor piston only a little over three times, instead of six times, the mean belt power.

From these diagrams it would appear that with belt-driving a small single-acting machine requires more fly-wheel

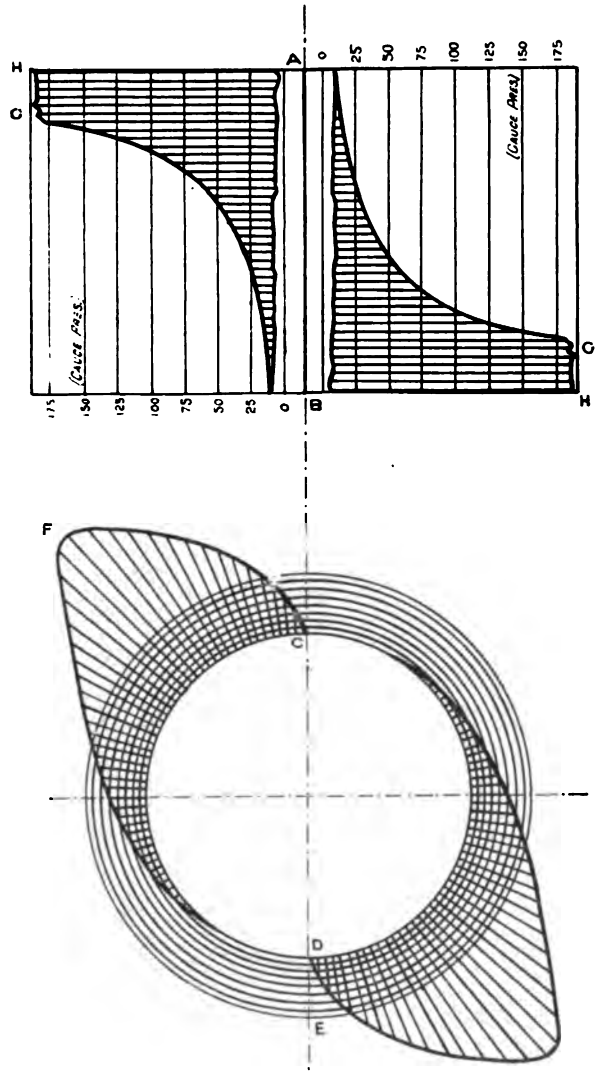


FIG. 99.--DIAGRAM TWO SINGLE-ACTING BELTED COMPRESSORS.

than one of twice the capacity, if it is double-acting and has double the belt power.

Fig. 100 shows a belted compressor of the open type, as designed by the author for small power, where a submerged

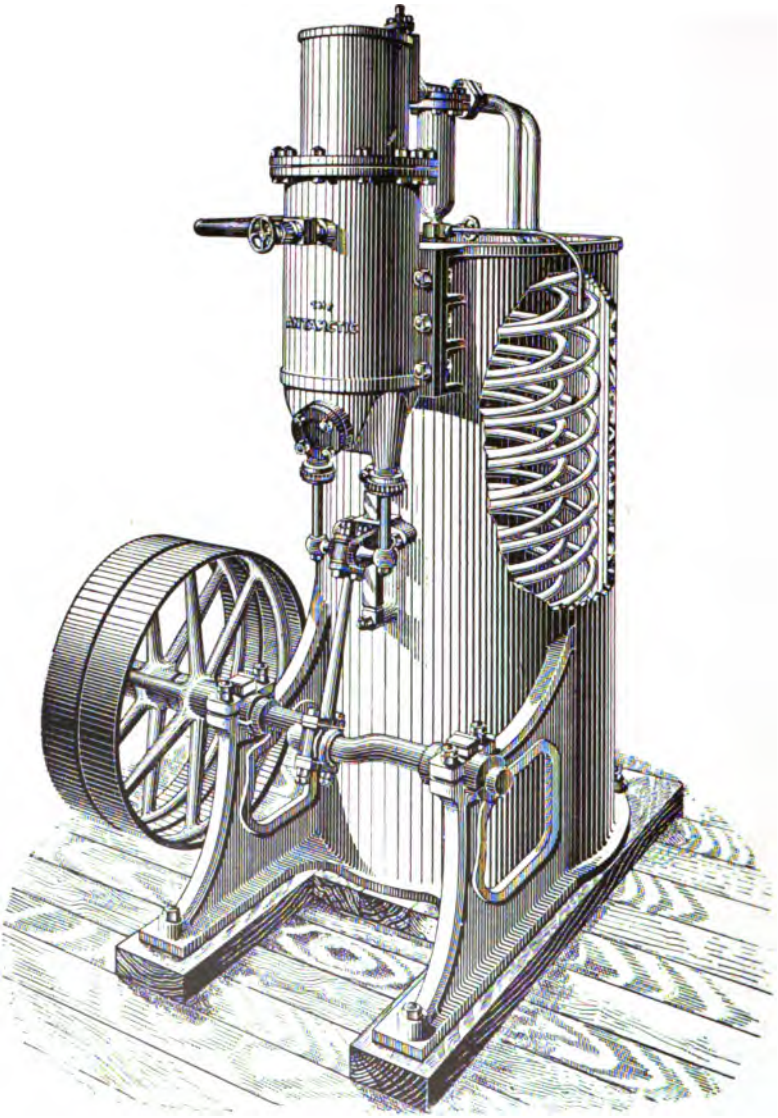


FIG. 100.—BELTED COMPRESSOR ON SUBMERGED CONDENSER.

condenser is preferable. This is an extremely simple machine, although the compressor being compound, the

framework and condenser tank are in one casting, and carry the single crank with overhung belt pulleys. Very little or no fly-wheel is necessary with this machine on account of the equable turning moments, which is described fully with diagrams under the head of compound compressors.

Fig. 101 represents two similar compressors coupled together and driven by disc cranks on a straight shaft; each of these is double the power of that shown by Fig. 100.

By "closed" machines, is meant all those in which the crank and connecting rods, which give motion to the compressor pistons, are inclosed in a chamber connected with the gas inlet, and so subjected to the back pressure. There are no piston rod packings required in such cases, and the main stuffing-box is around the crank shaft, where the packing is subjected to a slow rotary wear, which is continuous and in one direction, instead of to the more rapid and reciprocating wear of the ordinary piston rod. If the oil level is maintained above the top of the shaft in these machines, all escape of gas is prevented through the packing and glands.

The Westinghouse machine, Fig. 57, is one of the finest examples of this class of machine that is built; the builders say they believe in putting their eggs in a number of baskets, and prefer small units as more economical where the work varies with the season. Although the jointing of the back cover of these compressors with a simple flat surface, which requires ports to be cut in the jointing material, has been referred to as an undesirable feature, and although horizontal valves are not so trustworthy as vertical ones, yet there are in this machine a number of points which commend themselves to the experienced engineer, and evidence careful thought. Among these are the plain barrels or liners to the compressor cylinders; these enable hard and homogeneous metal to be used, and permit of simple renewals. The disposition of the centers of the cylinders above and below the center of the yoke—so that when the crank shaft revolves in the proper direction the twisting strain on the yoke is practically neutralized—is a sound mechanical device. There have been some bad imitations of this machine seen by the author, where the true spirit of the original was quite lost.

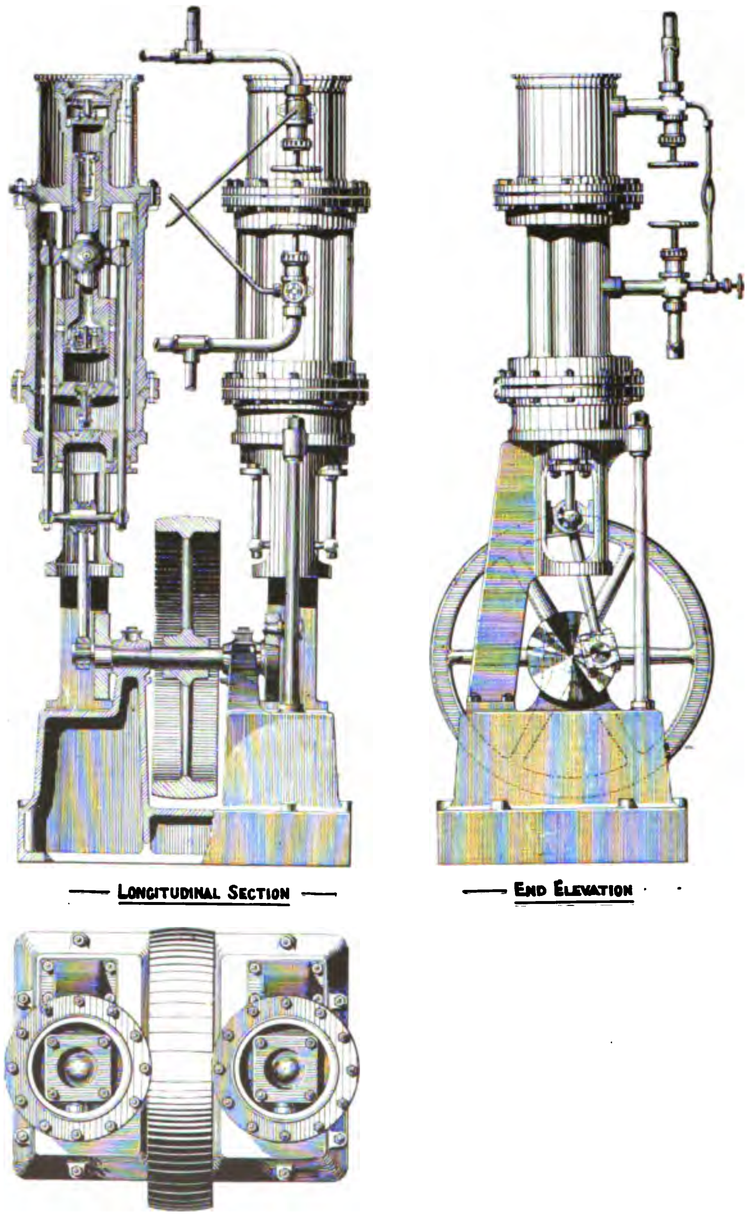


FIG. 101.– COMPOUND COMPRESSOR, TWO-TON ICE MAKING PLANT.

The Remington machine is representative of numbers of different makers' closed compressors which only differ from one another in small details; in all these cases there are one or two open-mouthed cylinders arranged over the crank casing, to which the return gas is led. The improved Remington machines are built with both suction and discharge valves in top head of the cylinder. For latest compressor see Chapter XX.

Fig. 102 is a section, and Fig. 103 a perspective view of an inclosed machine, with compound compression, having

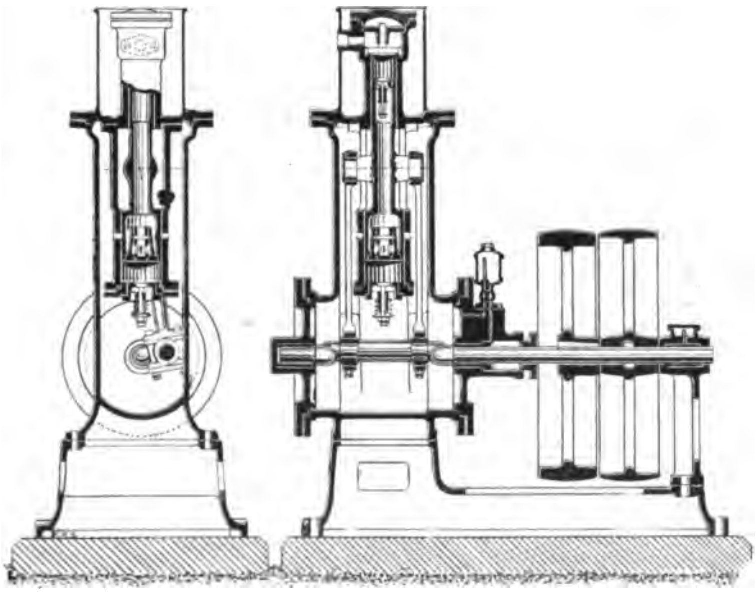


FIG. 102. SECTION OF ENCLOSED ANTARCTIC COMPRESSOR.

both cylinders opening directly into the crank chamber, and taking power on both the up and down strokes. The two connecting rods at their small ends are coupled direct to a crosshead on the trunk between the pistons. The details of this machine are described herein fully under compound compressors. In place of an internal crank shaft, some inclosed machines work by means of levers or beams inside the casing, and when the main center or vibrating axis which gives the motion is kept down in the bottom of the casing, a very small quantity of oil is sufficient to seal the shaft at the stuffing-box.

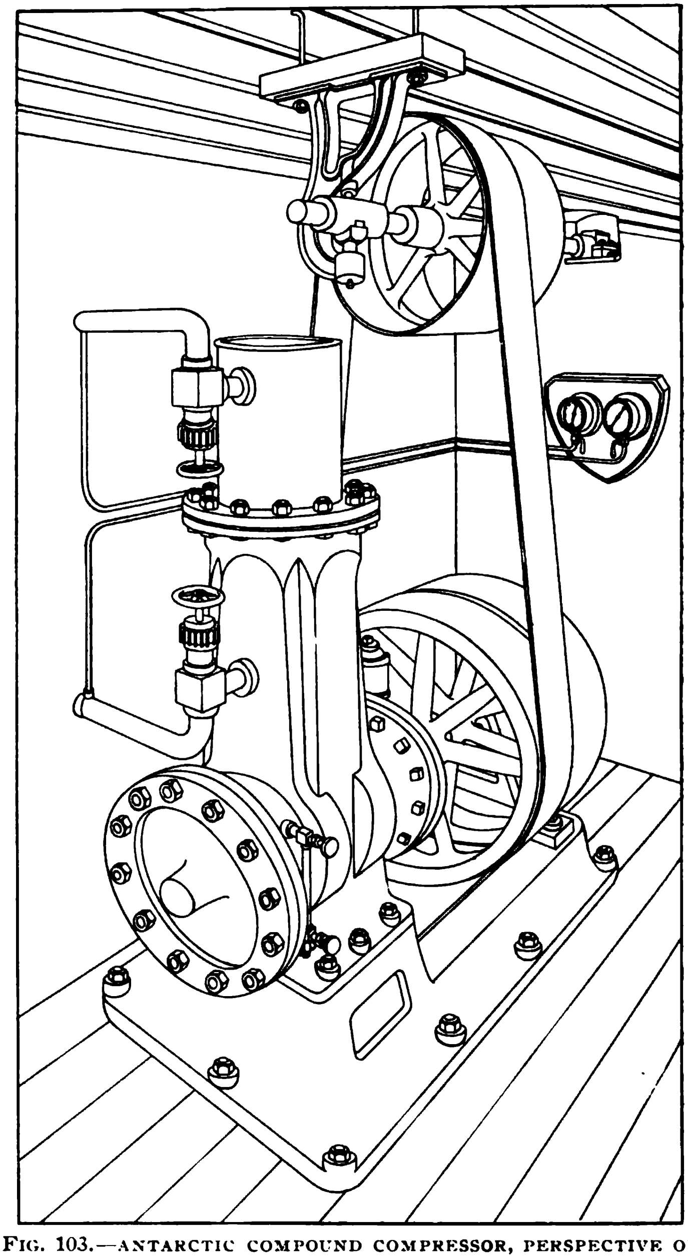


FIG. 103.—ANTARCTIC COMPOUND COMPRESSOR, PERSPECTIVE OF ENCLOSED TYPE MACHINE.

COMBINED MACHINES.

Small refrigerating machines are sometimes made not only with their condensers combined on one sole plate as in Fig. 100, but with both condenser and refrigerator all combined, as in the carbonic acid machines, Figs. 11 and 12. Fig. 104 shows two English machines of the Remington type with inclosed cranks, which are driven by a vertical intermediate engine; and with the condenser, refrigerator, and circulating brine pumps, are all erected complete on one sole plate. A

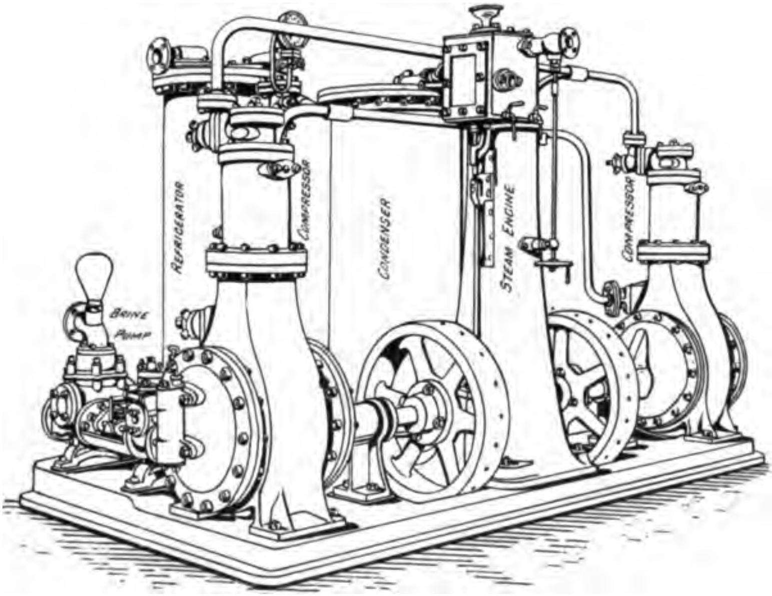


FIG. 104.—ENGLISH MACHINE, KILBOURN ENCLOSED TYPE.

number of these have been made by the Kilbourn company for shipboard use. Fig. 105 shows a step further than the last figure, and represents a small machine of one and one-half tons refrigerating capacity, with its boiler as well as a submerged condenser all fitted up complete—including its feed injector—on to one cast iron sole plate. This machine is made by the Clyde Engineering Co., Ltd., of Sydney, for special requirements.

COMPOUND AMMONIA COMPRESSORS.

The subject of compound compression has already been referred to once or twice on previous pages, but it is one

which should have fuller consideration given to it, because there is no doubt the system is making headway in connection with mechanical refrigeration; and it is possible that in the near future compound ammonia compressors will be the rule, as they are now the exception.

If it be said that compound compression complicates the machinery, and that the ice manufacturer or cold storage proprietor wants things as simple as possible, it will be well to show that there is not necessarily any more complication and there need be no greater number of parts with a compound compressor than there is with a pair of ordinary single-acting compressors; and that under some arrangements the compound machine is really much the cheaper, simpler, and better in the matters of first cost, multiplicity of parts, and the attention required when at work.

Admitting for the sake of argument that a compound compressor actually has the same number of working parts as an ordinary double-acting, or a pair of single-acting compressors, let us ask: What are the inducements to lead to its adoption? The answer to this is: First, a great equalizing of the turning moments, which lessens the loads or strains on the cranks, connecting rods, and pins. This enables these parts to be made of less strength, and so reduces the cost of the machine, while the lessened friction of the wearing surfaces economizes the power required to drive it. Secondly, the ability to cool the gas in the intermediate stage, which—by reducing its volume—enables the work of the engine to be lessened to a most important extent in large installations; and, Thirdly, the much smoother working, and reduced wear and tear in the whole machine, which is secured by the altered conditions.

The idea of compounding a compressor appears to be more than thirty years old, as there were patents granted in connection with it in 1867. The first compound ammonia compressors in Australia were built in 1885 for the Fresh Food and Ice Co., of Sydney, to the designs of their chief engineer, the late W. G. Lock, who patented a special invention to maintain the space below the pistons, in both high and low compressors, at the back, or refrigerator, gas pressure.

Among the notable builders of compound ammonia com-

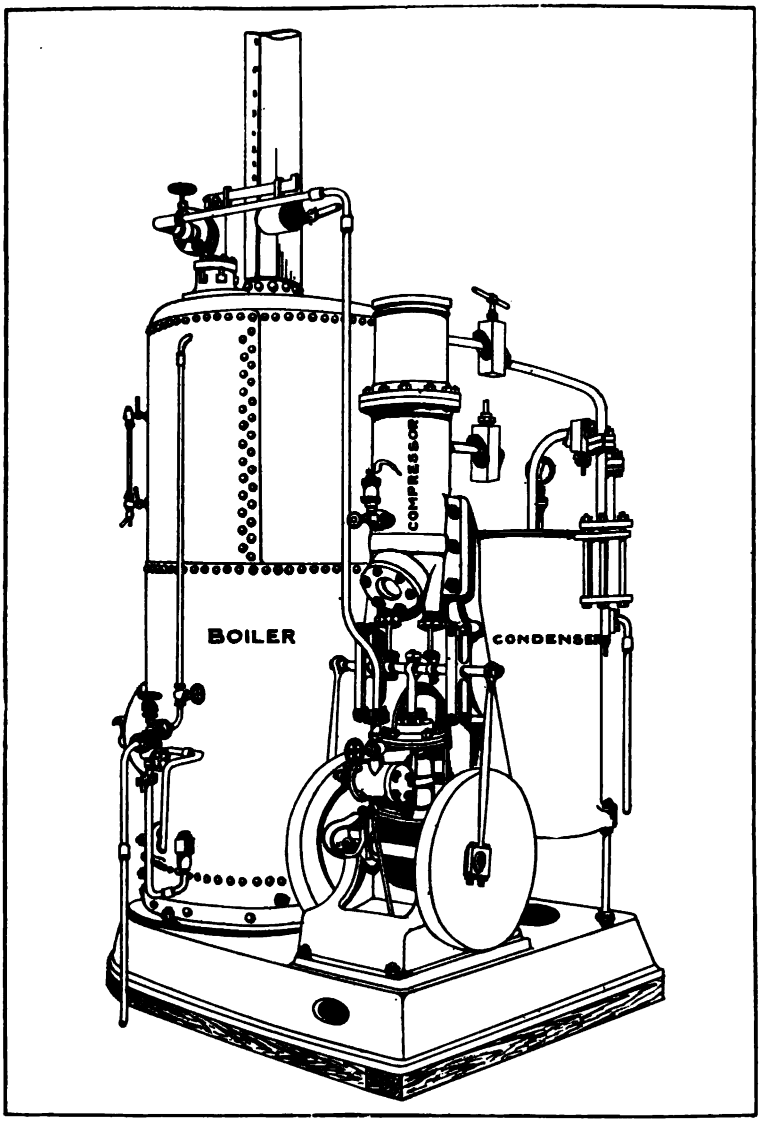


FIG. 105.—SMALL ICE MACHINE—CLYDE ENGINEERING CO., LTD., NEW SOUTH WALES, AUSTRALIA.

pressors at the present day there are The York Co., of York, Pa., U. S. A.; The Linde Co., and The Haslam Co., in England; Clyde Engineering Co., Ltd., in Sydney, N. S. W., and Humble & Nicholson, of Geelong, Victoria.

Fig. 70, on page 115, represents a 35-ton ice machine by the York Co., and Fig. 66 is a section of the two cylinders of the compressor by the same makers, who always arrange them vertically, while the steam engine may be either vertical or horizontal. In the case of only one high and one low pressure cylinder, the arrangement of the whole machine may be the same as in either of the Figures 81 to 85, with horizontal engines; or as 86 and 89, with vertical engines. Larger machines, as shown by Fig. 70, are made with two low pressure cylinders, which are on the outside, with the one high pressure cylinder between them; in this case the crank pins of the low pressure pistons are in line together, and the high pressure crank pin, on which the horizontal engine works, is at 180 degrees—or opposite. Under a design for a still more powerful machine, the builders place four compressor cylinders in a row, operated by four cranks; two being high pressure, on the outside, with the low pressure cylinders between them; and the two connecting rods, from a cross-over compound engine, operate one each on the two outer cranks. There are here *five* main bearings on the shaft, *four* crank pins, and *four* crosshead pins to look after; the fly-wheels are overhung, and therefore must wear down the outer bearings more than the inner ones. (All that is effected here can be carried out with two bearings, two cranks and one fly-wheel.) The pipes from the cylinder heads connecting the high and low pressure cylinders, through the intervention of the intermediate condenser or cooler, are seen at the top of the machine in Fig. 70. (For latest design see Chapter XX.)

Messrs. Humble & Nicholson, of Geelong, make great numbers of small compound ammonia machines for the butter and cheese factories of Victoria, which have two single-acting compressors arranged side-by-side, driven by a pair of cranks set at 180 degrees; but their machines are all horizontal instead of vertical like the York machine.

There is a peculiar feature about all these single-acting side-by-side compound compressors, in that the smaller or

high pressure piston is actually a motor on the "down" or "out" stroke. When the gas is being expelled from the large cylinder into the small one, it is of course compressed into the smaller volume at an increasing pressure, and this pressure acting on the smaller piston constitutes it a motor, which, acting on the high pressure crank, assists the rotation of the shaft, and therefore assists the engine to force up the large piston against its increasingload.

If there was no friction this would mean that the work which the engine had to perform would be equivalent on that stroke—that is the low pressure or first compression—simply to the pressure of the gas on the difference in the areas of the two pistons. Unfortunately, however, in such cases there is a great deal of friction, and with such machines, the work which the small piston is theoretically able to do is materially discounted, because it has to be transmitted through two pistons, rod packings, two crosshead pins and guide blocks, two crank pins, and the main bearings of the shaft; with friction upon friction, causing increased wear and tear, demanding more attention, and resulting in loss of power at every transfer.

In the compound ammonia compressors made by the Linde and Haslam companies, this loss of power and wear and tear are avoided, because the high and low pressure pistons are both coupled on to one piston rod, and the cylinders are connected by an intermediate chamber in connection with the suction or back pressure side. Under this arrangement the pressure of the gas in its intermediate stage is conveyed by a pipe from the front of the low pressure piston to the back end of the smaller cylinder, where, acting on the smaller piston in the opposite direction, it directly, and not indirectly, balances an equivalent area on the large piston. In such case, of course, the transference of power is without any friction or wear and tear due to journals and brasses, as in the other plan; and it certainly is a much better mechanical arrangement from an engineering standpoint.

Fig. 106 shows such a Linde compound ammonia compressor, of European make, combined with a compound steam engine. In this machine the whole engine power is applied to one crank, the whole of the power to work the

compressor being taken off the other crank. In this there is a considerable amount of friction, and a relatively very strong shaft is required, with bearings to correspond, to withstand the combined strains, or the sum of the working stresses of the two machines.

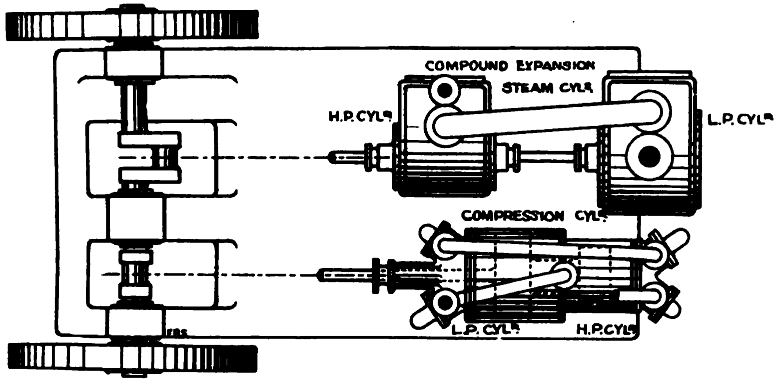


FIG. 106.—COMPOUND ENGINE AND COMPOUND COMPRESSOR (LINDE).

The "Antarctic" compound compressor is so designed that the pressure of the gas during the filling of the smaller cylinder acts upon its piston and directly balances an equivalent area of the large piston, just the same as in the Linde machine (Fig. 106); but the whole arrangement is simplified by the device of casting the two pistons together, and then passing the gas through the center of them both, from the low to the high pressure cylinder, instead of conveying it around by a circuitous route of pipes and passages.

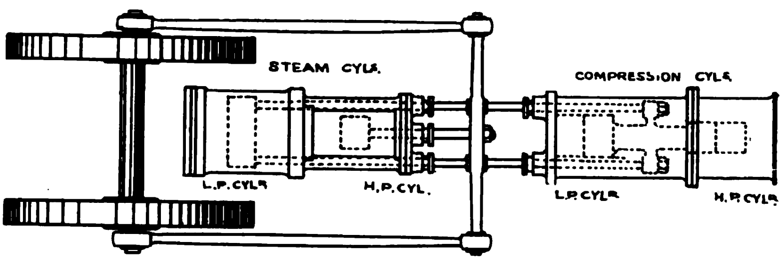


FIG. 107.—COMPOUND ENGINE AND COMPOUND COMPRESSOR (ANTARCTIC).

Fig. 107 shows one of these machines combined with a compound engine, where nearly all the work is communicated directly from the engine to the compressor, and the crank shaft and cranks only take up the difference, instead

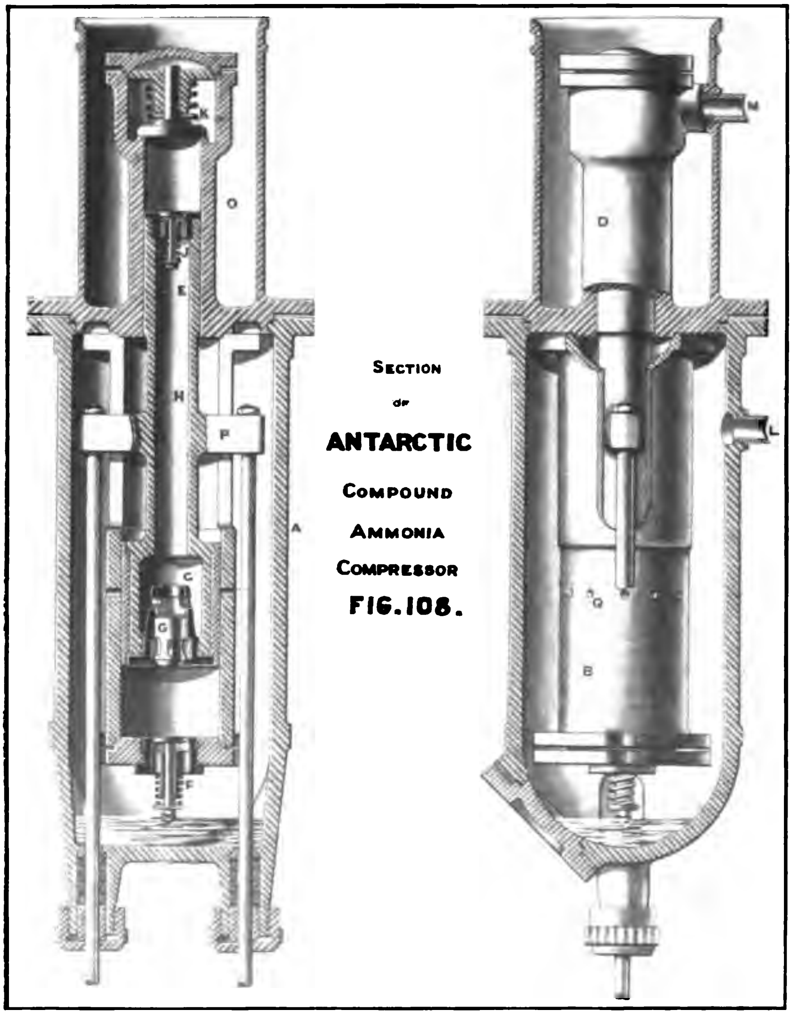
of the sum, of the loads. If this arrangement is compared carefully with Fig. 106 it will be seen that in the latter case the work of the connecting rods and crank shaft is very much less.

Fig. 108 is a section through the enclosing casing and two cylinders of the "Antarctic" Australian compressor (see following page), with the following explanatory references:

A. Main casing enclosing the two cylinders.
B. Low pressure cylinder.
C. Low pressure piston.
D. High pressure cylinder.
E. High pressure piston.
F. Low pressure inlet valve.
G. Low pressure outlet valve.
H. Passage from low to high pressure cylinder.
J. High pressure inlet valve.
K. High pressure outlet valve.
L. Main inlet branch.
M. Main delivery branch.
N N. Piston rods.
O. Water jacket to h. p. cylinder.
P. Crosshead to piston trunk.
Q. Openings to insure the filling of cylinder at full back pressure.

As the two cylinders both open into the casing, any leakage past the pistons is intercepted, and as there are no piston rods attached to the centers of the pistons, very large central valves can be fitted in. The top cover or bonnet is secured by only four large bolts, and when the four nuts are off, three of the valves are accessible, as the valve in the low pressure piston is so made as to come right up through the trunk. The lower valve is accessible by means of the special door, which also enables the casing to be cleaned. As the rods do not work through the cylinder bottom, they can be sealed with a considerable depth of oil in this casing without any risk of it being drawn into the system. The cylinders are plain barrels or pipes, and thus they can be easily cast, bored, lapped and renewed. As the valves in the pistons open during the down stroke, and insure practically equal pressure in the two cylinders, the work to be done on the down stroke is found by simply taking the mean *intermediate* pressure, less the *back* or casing pressure from the refrigerator, and multiplying it by the area of the annulus of the large piston. The upper annulus is of course always exposed to the casing pressure. With the relative areas of the pistons

at 3 : 1 the resistance or load on the down stroke is then the *mean pressure as above*, multiplied by *two-thirds* the area of the low pressure piston, and for the up stroke, the load is



SECTION OF ANTARCTIC COMPOUND AMMONIA COMPRESSOR FIG. 108.

the *forward pressure less the casing pressure*, multiplied by only *one-third* the area of the large piston.

It will be as well in order to facilitate a proper comparison between an ordinary single-acting compressor, and one

of the type illustrated by Figs. 106, 107 and 108, to assume a certain size of machine and then calculate the loads on the two pistons, and the strains or stresses on their respective crank pins, connecting rods, and other parts. Taking therefore as a very common size a 20-ton refrigerating machine, we may assume a compressor diameter of a little over eleven inches, or say for round numbers, 100 square inches, as the area of the piston, with a back pressure of twenty pounds by the gauge, or thirty-five pounds absolute, and a condenser pressure of 160 pounds by the gauge, equal to 175 pounds absolute, then the ratio of compression would be 5 : 1.

In the ordinary compressor, the load on the piston after the full pressure is reached will be 175−35=140 pounds × 100 =14,000 pounds. This pressure continues to the end of the stroke, and all the parts of the machine must be proportioned for this amount of stress or load.

In the case of the compound compressor, as Fig. 108, with the small piston one-third the area of the large one, the area of 100 square inches would have one-third or 33.3 square inches of it neutralized by the pressure on the piston above, as it is manifestly the same pressure in both cylinders during the down stroke. Hence the effective area of the large piston acting on the gas being compressed would be 66.6 square inches only, instead of 100 square inches. If the gas is humid, or is compressed in accordance with Mariotte's law, isothermally, into one-third of its original volume, the pressure will rise to 35×3=105 pounds absolute, in the first compression, and if compressed without loss of heat, or in accordance with the adiabatic law, it will reach to about 114.7 pounds absolute. Assuming then a dry compression, the maximum resistance to the low pressure compound piston will be 114.7 pounds, for round figures, say 115−35=80 ×66.6=5,328 pounds, which is the greatest stress or load on the machine, instead of 14,000 pounds, as in the other case. Truly a wonderful reduction in the strains to be provided for, in designing shafts, rods and bearings.

In the final compression or up stroke the load cannot be more than 175—35=140 pounds by 33.3 square inches=4,662 pounds. This is a less final pressure than the down stroke, but the point of expulsion is reached much earlier, so that in

practice the horse powers of the up and down strokes correspond very closely.

It is abundantly clear from the foregoing calculation that the load or stress on the motion gearing of a simple compressor is from two and one-half to three times as great as it need be in one of these compound compressors of equal capacity, quite apart from the disadvantages of unequal running, trouble about clearance, and limited piston speed possible, which do not apply with the same force to compound compressors. If this has no more significance than the fact that the same weight of crank shaft, crank connecting rods, and such gearing, which is necessary for an ordinary compressor of twenty tons capacity, will answer for a compound compressor of fifty tons refrigerating power, it is even then sufficiently startling to inspire the inquiry: If this is true why are compound compressors not more commonly used?

No doubt the scoffer will say, "I could make my compressor double-acting and then I would only have 7,000 pounds, not much more than your 5,328 pounds," but he would have full pressure at both ends, loss by clearance and short period of expulsion—all absent from the compound machine.

Incidentally, there is another feature in the compound compressor which has advantages, in that it produces a more continuous flow of gas from the refrigerating coils, approximating to ordinary double action. The effect of the large piston on the *down stroke* is to draw into the casing *two-thirds of the low pressure cylinder-full* from the refrigerator, owing to the enlargement of the capacity of the casing chamber by that volume. On the up stroke this *two-thirds* is put back into the chamber, and *three-thirds*, or full volume, is drawn in at the bottom inlet valve; consequently the balance, or one-third of cylinder's volume, is drawn into the casing on the *up stroke*. This double flow of gas into the casing reduces the friction on the inlet or suction pipe.

As the result of the equable distribution of the work throughout both the up and down strokes, these compressors run very steadily, and the author saw one making 140 revolutions a minute, as it stood on the fitting shop floor, without

a single holding-down bolt. It was found to be quite steady at that speed.

In simple compression, with the forward and backward gauge pressures at twenty pounds and 140 pounds, or say thirty-five pounds and 155 pounds absolute, the ratio of compression would be about 4½ : 1, and the gas would all have to be expelled through the delivery valve into the condenser during the short period of, say, one-fourth of the piston's stroke. With the compound machine a 3 : 1 compression would already exist when the second compression began, so it is evident, as $\frac{3}{2}\times\frac{3}{1} = 4\frac{1}{2}$, that when the high pressure piston

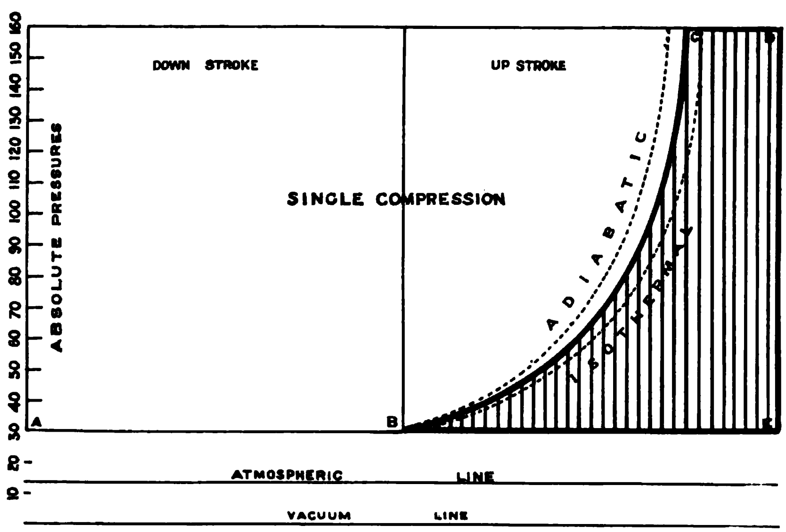


FIG. 109.—THEORETICAL DIAGRAM SINGLE-ACTING COMPRESSOR.

has traveled one-third of its stroke the terminal pressure will be reached, and therefore the expulsion of the gas would be distributed over two-thirds of the stroke instead of being all concentrated on the last quarter of the stroke. This proportion is as eight to three, therefore the compound delivery would extend over more than two and a half times as much of the piston's stroke as the other one would do. With this free get-away and more uniform delivery, there is less banking up of pressure and oscillation of the pressure gauge indicator, by the friction of the pipes and the jerky supply to the condenser.

Figs. 109 to 112 illustrate graphically the stresses that have just been described, and some of the special features of compound compression by theoretical diagrams. Fig. 109 is an indicator card from an ammonia compressor, working between thirty pounds and 160 pounds pressure absolute. From A to B, is the down stroke on which no work is done, from B to C shows the work of compression, and C to D the period of expulsion. Both isothermal and adiabatic lines are shown, and the actual curve is taken for the purpose of comparison, half way between the two.

In Fig. 110 the diagram of the first compression to one-third of the original volume, shows the isothermal line at

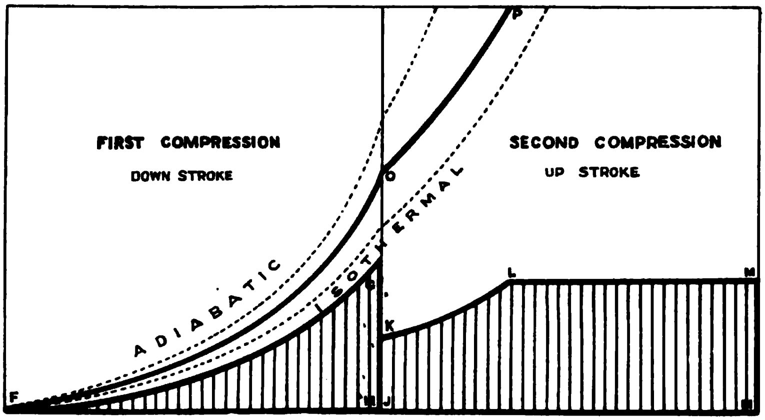


FIG. 110.—THEORETICAL DIAGRAM ANTARCTIC COMPRESSOR.

$30 \times 3 = 90$ pounds, the adiabatic curve rising to over 120 pounds, and the mean pressure at the point O, about 107 pounds. In the second stage of compression, carried on in the smaller cylinder, the curve reaches expulsion pressure at the point P, or about one-third of the stroke.

In order to ascertain the effect on the piston rods and crossheads of these different gas pressures the full piston area for the down stroke must be considered as reduced by one-third, and for the up stroke by two-thirds, which gives the two points G and K on the card as the result of first compression. The point L corresponds with P, and thus while the line E O P in Fig. 110 corresponds with B C in Fig. 109 so far as gas pressure on the square inch goes, the spaces

which are hatched with vertical lines represent in proper proportion the actual relative amount of work performed by the several pistons, and the different lengths of the vertical lines, the relative stresses or resistances to which the piston rods are subjected under the two systems, with the actual work of same in both cases.

In these diagrams the greater proportion of the stroke L M during which expulsion takes place in the small high pressure cylinder is very noticeable when compared with C D in the simple compressor. It must be understood that J K

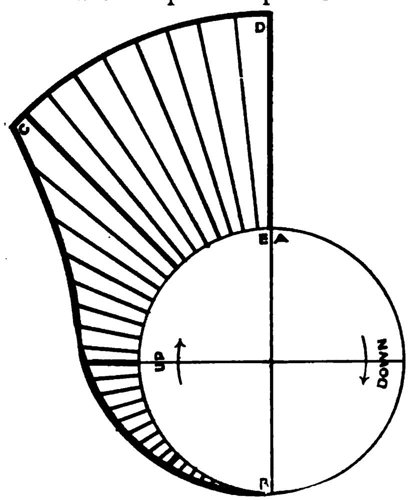


FIG. 111.

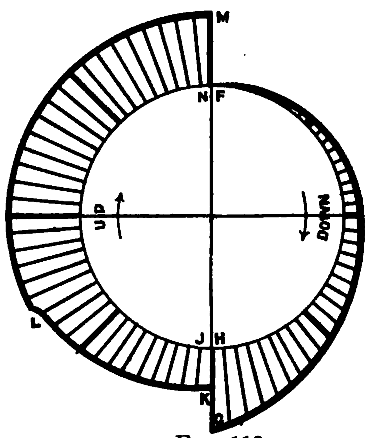


FIG. 112.

and H G both represent the same or intermediate pressure as the greater length J O. The area of the simple cylinder being taken as unity, is represented by J O. The low pressure compound cylinder's effective area being two-thirds of unity, H G is two-thirds of J O, and similarly the high pressure cylinder being one-third the area, J K is drawn one-third of the height, to show graphically the absolute pressure on the whole piston, instead of the pressure per square inch.

Figs. 111 and 112 almost explain themselves. They are constructed by simply curving Figs. 109 and 110 round into a circle; they exhibit the almost steam hammer action of the

one case, as compared with the even distribution of the load in the other and compound one, Fig. 112.

Figs. 113 and 114 are copies of actual indicator cards taken with separate springs from an Antarctic compressor

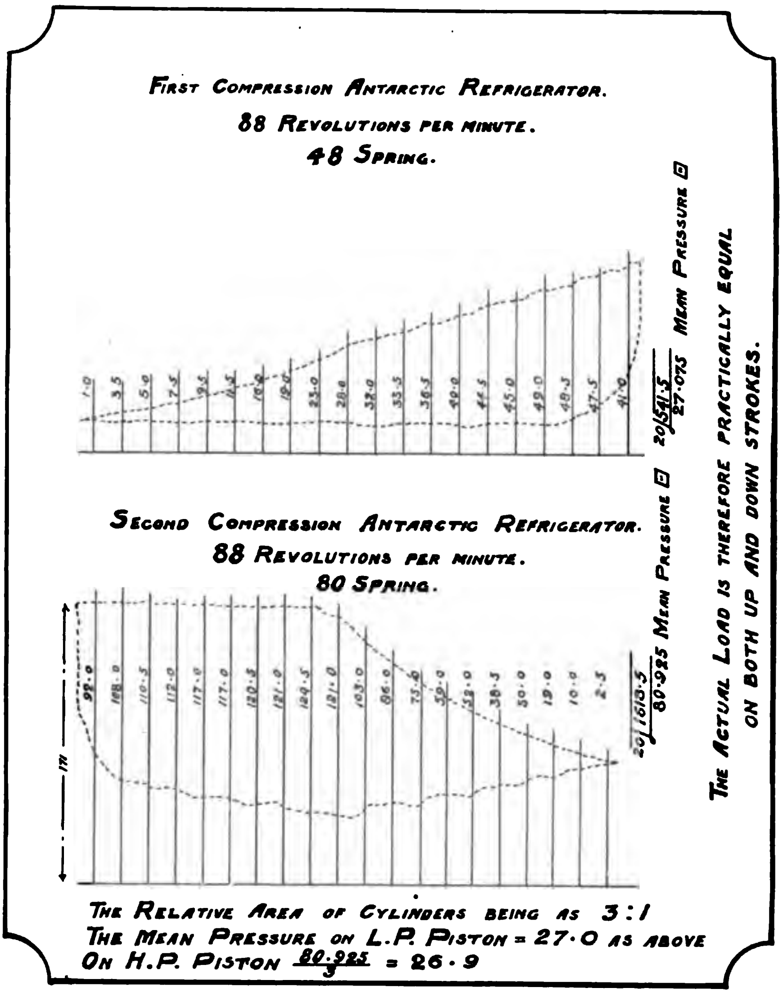


FIG. 113.—INDICATOR CARDS FROM AMMONIA COMPRESSOR.

running at eighty-eight revolutions, and up to 171 pounds pressure absolute. In constructing the theoretical cards, Fig. 110, no account was taken of the chamber H, shown in

Fig. 108, as it does not affect the ultimate results, but its effect on the actual indicator card is clearly seen.

With isothermal compression directly from a low pressure to a high pressure cylinder, and the pistons moving uniformly together, the low pressure diagram would be a triangle, and the line of pressures would be straight instead of a hyperbolic curve. With a chamber like H, in Fig. 108, which on the completion of the down stroke is filled with gas of the same pressure as that in the high pressure cylinder, the lines are considerably altered, because no delivery from the low pressure cylinder will take place on the down stroke until equilibrium is established on both sides of its outlet valve; and this is kept shut by the greater pressure above it at the commencement of the stroke. The inclosed gas, however, is free to pass through the upper valve into the small cylinder, and consequently when the pistons descend it enters freely into the same, the pressure falling above and rising below until the two cylinders and the connecting trunk are in equilibrium; when such is the case, the lower valve opens, and the high and low pressure cylinders are then in direct communication with one another.

In Fig. 115 the two cards are reduced to a common scale and plotted together like the cards from compound engines, such as the Westinghouse, and show as follows: Commencing the down stroke at A, from A to B the gas is being compressed in the one cylinder alone, and the line is so far the ordinary curve, during which time the pressure in the upper cylinder has been descending from D to E by the expansion of the entrapped gas in the chamber into the small cylinder, when equilibrium is established. This equilibrium is shown at the points B, on the low pressure card, and E, on the high pressure one. From B to C the gas is passing from the low to the high pressure cylinder, and the line E to F is practically straight, corresponding with the line B to C, except so far as it is affected by the resistance of the valves or their springs. The curve F to G is the line of final compression, and the distance G to H the period of expulsion to the condenser. The curve at J is due to the delay in the opening of the admission valve to the low pressure cylinder.

Notwithstanding the much higher pressure— say 1,400 to 1,500 pounds to the inch—at which carbonic acid machines are worked, and although air is now compressed to thousands of pounds pressure every day (for torpedo work, and so on), the author has not yet met with an example of a com-

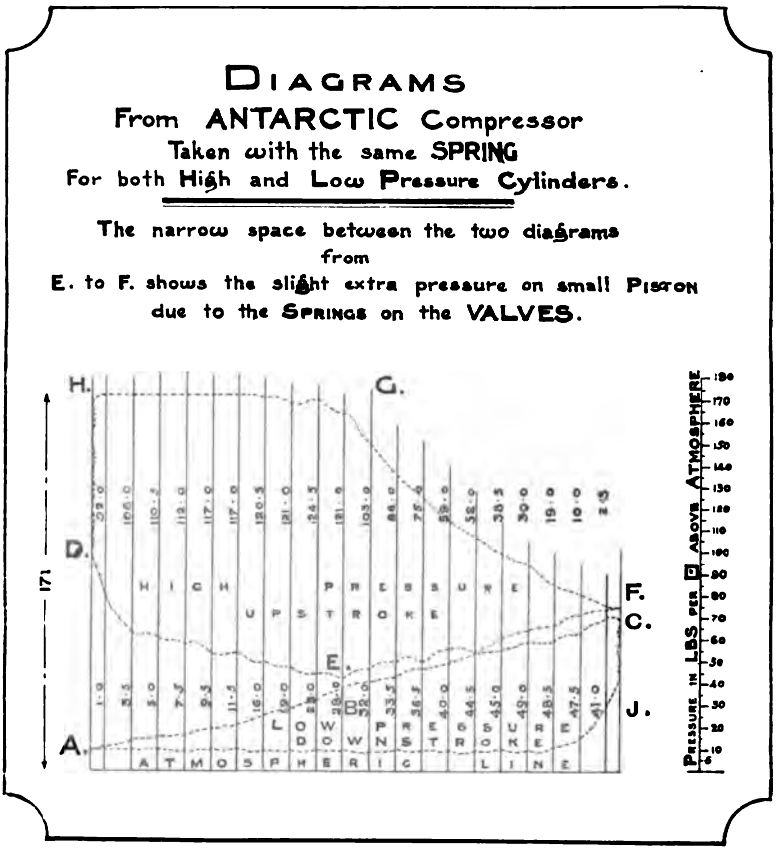


FIG. 115. INDICATOR CARDS FROM AMMONIA COMPRESSOR.

pound carbonic acid machine. This is probably because owing to the high back pressure the ratio of compression in such machines is less than with ammonia compressors, but there certainly seems no reason why they should not be used, if only to save the trouble and loss occasioned by leakage at

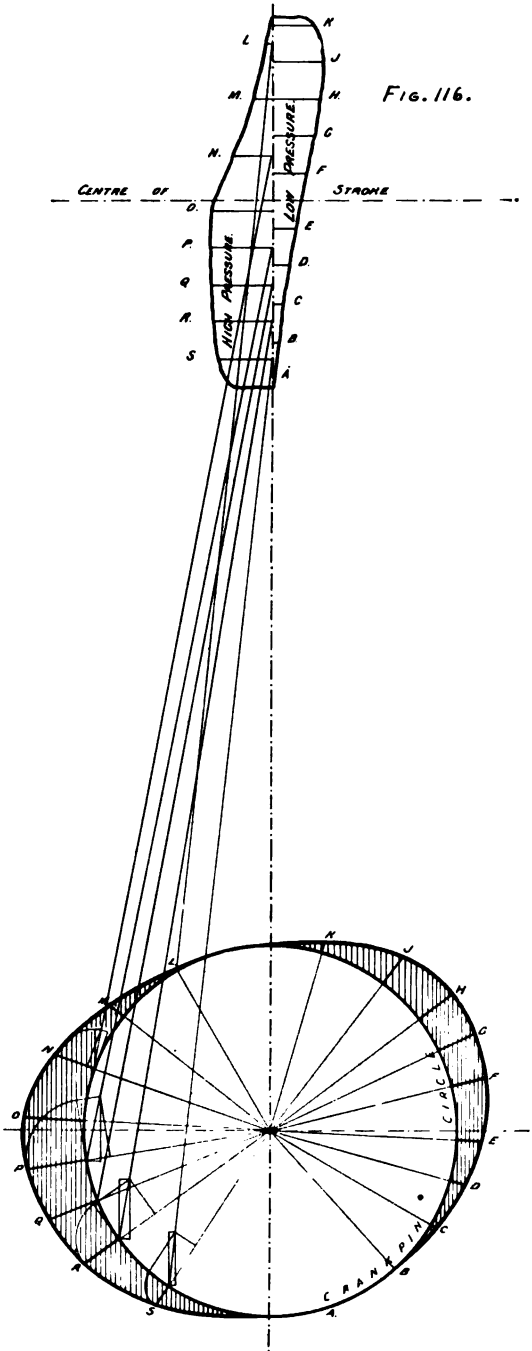


FIG. 116.—DIAGRAM SHOWING THE WORK ON PULLEY, OR TURNING MOMENTS, WITH COMPOUND COMPRESSOR.

the piston rod packing which has to stand the full forward pressure.

Fig. 116 shows, by contrast with Figs. 98 and 99, the great advantages possessed by compound compressors in securing equable turning moments, particularly when they are belt driven. The upper diagrams on the Fig. 116 are identically the same as those shown in Fig. 115; but corrected for relative areas for the pistons, so as to show relative absolute pressures on the whole piston's areas, instead of pressures per square inch. To prevent the confusion of a great number of small overlapping lines, the graphic construction is given for four positions only of the connecting

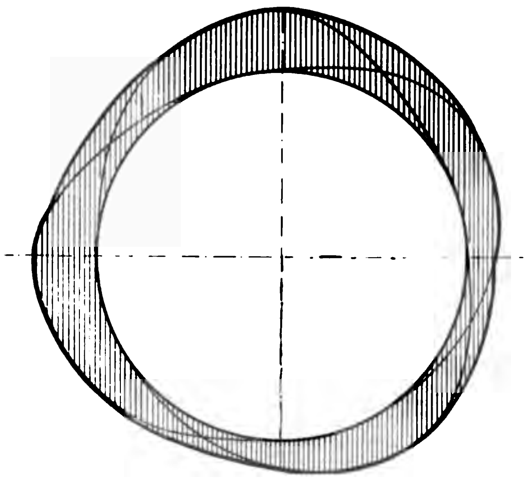


FIG. 117.—TURNING MOMENTS WITH TWO COMPOUND COMPRESSORS.

rod, which is made two-and-a-half times the length of the stroke. By following the several parallelograms of force, it will be seen that the height of the ordinate representing the pressure on the piston, is first resolved into the horizontal pressure on the compressor guide, and the force acting in the direction of the center line of the connecting rod. This force acting on the connecting rod, is then resolved into radial thrust or pressure on the crank pin and shaft, and the tangential force acting on the crank pin. The radial ordinates set up outside the circle are the same length as those on the tangents. The force or pull of the belt which directly

corresponds with the power represented in the cylinder diagrams is, therefore, shown in all positions of the crank or pistons by the curved outer line joining the radial ordinates, and its distance from the crank pin circle. This illustration applies specially to the machine shown by Fig. 100.

When two compound compressors are coupled with their cranks, at right angles, the resulting turning moments — as shown by Fig. 117 — are so uniform, that it is evident no fly-wheel whatever would be required if there was any margin of belt and pulley power. This diagram applies particularly to compound comprèssors of the general type shown by Fig. 101, modifications of which are also made by the Linde and Haslam companies.

CHAPTER XVI.

ON THE LAWS RELATING TO THE EXPANSION AND COMPRESSION OF GASES.

The acquirement of a thorough familiarity with the laws which govern the compression and expansion of gases, can hardly present any difficulty, at the present day, to the college trained youth or university student. The education of such persons should put them in touch with mathematical text books, which now make more or less reference to that branch of knowledge, and whole volumes are to be found, which have been written for their instruction in thermodynamic lore. The average refrigeration engineer, however, is, for many reasons, not always able to follow the intricate formulæ and equations with which such works abound. This chapter therefore has been written by a practical man (who knows more of the drawing office, machine shop and factory, than he does of the college class room), as an attempt to present to other practical men like himself, some information connected with the laws of gases, in a more simple form. It is hoped that this will not only assist such persons to investigate the operation of a compressor, but will also enable them to construct theoretical diagrams of the work that should be performed by its piston, so that they may compare them with the actual results, as given by the indicator.

If before entering upon this subject it is asked: What is meant by a gas? The reply is: The most distinguishing characteristic of any gas is its elasticity, or its capacity for infinite expansion. Not many years since there were many gases called permanent, which were supposed to admit also

of practically indefinite compression; but, although they are now easily liquefied by pressure, when cooled below their critical temperatures, no practical limit yet exists to their expansion. It is found that as the pressure on any gas is diminished, so its volume increases, and that, before all the pressure could be removed, the volume would become so great, that no vessel could be found large enough to contain it.*

As a consequent result, when the temperature of any gas is increased, either the pressure or the volume, or both pressure and volume, will increase also, and these operations follow more or less closely certain laws, which are generally known as the laws of gases.

Our knowledge of these laws is based on the researches of the past two hundred years, and the greatest advances, or those which have led to their comprehension on mechanical, as distinguished from mathematical grounds, have been made during the present century.

The establishment of the mechanical equivalent of heat by Joule (under which 772 foot-pounds are accepted as the equivalent of a thermal unit), has enabled the deductions from the earlier discoveries to be corroborated by a separate process of investigation.

BOYLE'S LAW.

The first law to be discovered and given to the world in connection with gases, is known indifferently as "Boyle's" law, or "Mariotte's" law. It was published by Robert Boyle in 1662, and Mariotte fourteen years later, in 1675, set it forth carefully verified in his treatise "De la Nature de l'Air." As French and other continental writers generally give the credit to Mariotte, English speaking people only do justice to the original discoverer, when they know it as Boyle's law.

Under this law, with any mass of gas at constant temperature, the product of its volume and pressure is a constant. Put in other words, in whatever proportion the pres-

*This may be better realized perhaps, if it is borne in mind, that the atmosphere, owing to this expansive property, extends hundreds of miles out into space, from the surface of the earth.

sure of a gas is to be increased, in just such proportion must its volume be diminished, or *vice versa*, the temperature in both cases remaining constant.

CHARLES' LAW.

The second law is called the law of Charles, after M. Charles, who was professor of physics at Paris, and who died in 1823. He is said to have been the first discoverer, although particulars of it were published by Dalton in 1801, and by Gay-Lussac in 1802.

Under this law, with a unit mass of gas under constant pressure, the volume increases from the freezing to the boiling temperature of water, directly as the temperature increases.

It has been found by careful experiment that air under constant pressure increases in volume, as it is raised in temperature from the freezing to the boiling points of water, in the ratio of 1:1.3665; or in other words, that 30 cubic inches or feet will increase to about 41 inches or feet, with such an accession of temperature.

It follows from this that if Charles' law is a correct one, within the limited range of his experiments, and Boyle's law is good for all temperatures, then Charles' law must also be true for other temperatures and pressures. Because, if—

Volume is denoted by	V,
Pressure by	P,
Temperature by	T,

Then Boyle's law says V P is constant when T is constant; but Charles says when P is constant, and V increases from 1 to 1.3665, then T rises 180°; therefore V P is increased at that particular pressure. But Boyle's V P does not depend on any particular pressure, and is true for all pressures. Hence, whatever the pressure of a gas may be, the product of the volume and the pressure, that is V P, will be increased in the proportion from 1 to 1.3665, by an increase of 180° Fahrenheit starting from the freezing point of water.

Experiments have been carried out with a great number of gases, and their expansion has been measured through the 180 degrees from the freezing to the boiling point, with the

result, that the maximum variation is found to range between 1 to 1.367 with air, and 1 to 1.390 with sulphurous acid. As a result of the researches of MM. Regnault and Rudberg, the ratio of expansion for the average gas, when raised from the freezing to the boiling point of water, may be taken as from 1 to 1.365. That is the volume increases 0.366, or 36.5 per cent, for an increase of temperature of 100° Centigrade, or 180° Fahrenheit; and, as the expansion or contraction is uniform with each degree, throughout the whole 180 degrees, then it is evident that the expansion for one degree will be $\frac{.365}{180}$ which is the same as $\frac{1}{493.2}$

This means that any volume of air at 32°, will expand or contract through $\frac{1}{493.2}$ part of its volume, for every Fahrenheit degree that its temperature is increased or reduced, if under uniform pressure throughout. Experimentally, this has been verified up to 700° above, and the law still been found to be true. Inferentially then it is assumed that air would necessarily contract in volume in the same way with a corresponding reduction of temperature, until arriving at 493.2° below the freezing point, or 461.2° below Fahrenheit zero, where it would be in a state of collapse without any remaining elasticity. This temperature of —461.2° is therefore called the absolute zero of temperature, and Fahrenheit zero is 461.2° of absolute temperature.

It will be understood from this that in order to double the volume of a given weight of air at 0° by the thermometer, it would have to be heated to 461°; and in order to treble its volume, to raise its temperature to 922°, and so on.

It must not however be inferred also, that these conditions apply absolutely and exactly to all gases, and under all conditions. Up to pressures as high as say 100 atmospheres, they appear to apply to the more permanent gases, such as oxygen and hydrogen, but not to the gases most used in refrigerating machines, such as ammonia, sulphurous acid, and carbonic acid; these are proved to be sensibly more compressible than air.

Carbonic acid under five atmospheres does not occupy more than 97 per cent of the volume which air would do under the same pressure, and under forty atmospheres,

near the condensing point, only 74 per cent, or less than three-quarters of the volume it should do on the basis given above.

Tables of the progressive pressures required to compress different gases have been published, one of which follows:

COMPRESSION OF GASES UNDER A CONSTANT TEMPERATURE.

(NOTE.—A meter of mercury equals 19.34 pounds per square inch.)

Ratio of the original to the reduced volume.	PRESSURES IN METERS OF MERCURY.			
	Air. Meters.	Nitrogen. Meters.	CO_2 Meters.	Hydrogen. Meters.
1	1.000	1.000	1.000	1.000
2	1.998	1.997	1.983	2.000
4	3.987	3.992	3.897	4.007
6	5.970	5.980	5.743	6.018
8	7.946	7.964	7.519	8.034
10	9.916	9.944	9.226	10.056
12	11.882	11.919	10.863	12.084
14	13.844	13.891	12.430	14.119
16	15.804	15.860	13.926	16.162
18	17.763	17.825	15.351	18.211
20	19.720	19.789	16.705	20.269

For the purpose of the practical calculations that are required in connection with every-day refrigeration, it should be sufficiently accurate to estimate on the assumption that the different substances which are used for the medium will behave as if they were perfect gases.

We can omit the decimal for convenience in ordinary calculations, and admit that a gas will increase $\frac{1}{493}$ part of its volume at the freezing point, or $\frac{1}{461}$ part of its volume at zero, for each degree increase of temperature. If then we have to deal with a given weight of gas at ordinary atmospheric temperature, say 65°, and desire to double its volume, it will not be sufficient to increase its temperature by 65°, and raise it to 130°. Such an addition is altogether beside the mark—the actual operation is as under:

In order to double 65°, first add 461°, which gives 526° absolute; and this multiplied by 2, equals 1,052°, absolute temperature. Deducting 461° gives 591°, as the thermometer temperature of the gas when its volume is doubled. Again $65^\circ+461^\circ=526\times3$ equals 1,578° absolute; deduct 461°,

gives 1,117°, as the Fahrenheit temperature, when its volume is trebled.

Therefore any current of air at 65°, such as is ordinarily supplied to the furnace of a steam boiler (where the pressure is practically constant), will occupy double volume at 591°, and treble volume at 1,117° of temperature.

Summing up all these several laws into a few brief sentences, it is found with gases:—

A. The pressure varies inversely as the volume when the temperature is constant (Boyle).

B. The pressure varies directly as the absolute temperature when the volume is constant (Charles).

C. The volume varies as the absolute temperature when the pressure is constant.

D. The product of the pressure and volume is proportional to the absolute temperature.

The pressure in all these cases is absolute pressure—measured from a vacuum; so that the atmospheric pressure must be added to that shown by an ordinary gauge, before making calculations, and the same must be deducted from calculated results, to give the gauge pressure.

The following simple rules based on the foregoing laws may be passed over by the advanced student:

1. With a known volume of a gas at any temperature (the pressure being constant), to find the volume at any other temperature. The sum is a simple one of proportion, V^1, P^1, and T^1, as before, standing for the volume, pressure and temperature unknown or required.

$V : V^1 :: T+461 : T^1+461$, and therefore—

$$V^1 = V\,\frac{T^1+461}{T+461}$$

2. With a known volume at a given pressure and constant temperature, to find the volume at any other pressure, then—

$$V^1 : V :: P : P^1 \text{ or } V^1 = V\frac{P}{P^1}$$

3. With a known volume, at a given pressure and temperature, to find the volume at any other pressure and tem-

perature. Here the operation is one of double or compound proportion, the result being in the compound ratio of the absolute temperature direct, and the pressure inversely; thus—

$$V : V^1 :: P^1\,(T+461) : P\,(T^1+461), \text{ or}$$

$$V^1 = V\,\frac{P\,(T^1+461)}{P^1\,(T+461)}$$

4. With a known volume at a given pressure and temperature, to find the pressure at any volume and temperature—

$$P : P^1 :: V^1\,(V+461) : V\,(T^1+461)$$

$$P^1 = P\,\frac{V\,(T^1+461)}{V^1\,(T+461)}$$

If the above equations are correct, and the volume of one pound of any particular gas at a given temperature and pressure is known, then it is evidently possible to find a co-efficient for such gas, which will save a great deal of trouble in making calculations connected with it. For instance, take air:

The volume of one pound of air at atmospheric density, or 14.7 pounds pressure to the square inch, and at 32°, is 12.387 cubic feet.

The absolute temperature is $32^\circ + 461^\circ = 493^\circ$, and hence—

$$\frac{12.387 \times 14.7}{493} = .36935 \quad \text{or} \quad \frac{1}{2.7074}$$

This fraction, 0.36935, is therefore a constant, which, when multiplied by the weight in pounds, and temperature of the gas in degrees absolute, and divided by the pressure in pounds per square inch, will give the volume in cubic feet; or conversely, the pressure at any volume in cubic feet, of one pound of air. Thus—

$$V\,P = \frac{461}{2.7074} \text{ for 1 lb. of air.}$$

The following table gives the value of this co-efficient (a) for six different gases, and any one of these values, multi-

plied by 144, gives the co-efficient (a) for pounds per square foot:

VOLUME, PRESSURE AND TEMPERATURE OF GASES.

Constants (a) for the equation $V P = a\ (T+461)$.

Name of gas.	Volume of one pound of gas at 32° F. under one atmosphere.	Value of the co-efficient (a).
Sulphuric ether............	4.79	0.1424 or $\frac{1}{7.019}$
Sulphurous acid...........	5.513	0.1643 or $\frac{1}{6.089}$
Carbonic acid..............	8.101	0.245 or $\frac{1}{4.1398}$
Air..........................	12.387	0.3693 or $\frac{1}{2.7074}$
Ammonia...................	21.017	0.6266 or $\frac{1}{1.596}$
Gaseous steam............	19.913	0.5937 or $\frac{1}{1.684}$

The volume of one pound of, say, ammoniacal gas, within ordinary working temperatures and pressures, is found as follows by the use of this co-efficient:—

$$V = \frac{T+461}{1.596\ P}$$

That is, take the weight of ammonia in pounds, multiply it by the absolute temperature, and divide it by 1.598 times the absolute pressure per square inch, to give you the volume in cubic feet.

To find the pressure of any volume of one pound of ammonia:—

$$P = \frac{T+461}{1.596\ V}$$

To find the density or weight in pounds of a cubic foot of ammonia at a given temperature and pressure:—

$$W = \frac{1.596\ P}{T+461}$$

THE SPECIFIC HEAT OF GASES.

In the earlier part of this volume, there is a table of the specific heats of a number of solid substances; these in all cases may be taken as constant quantities. M. Regnault is the authority for the assumption that the specific heat of a *given volume* of any one of the permanent gases is also practically constant for all temperatures and pressures, inasmuch as the variation through 360 degrees is not more than 0.2377. But gas has the property, which solids have not, of altering its volume considerably; and the specific heat of a gas

has to be considered from two separate points, not only when it is under constant volume, but also when it is under constant pressure.

The capacity for heat under constant pressure is much greater than under constant volume, and the comprehension of the reason for these two separate attributes of a gas will be much facilitated by the following diagram, Fig. 118.

Here we have a cylinder 35.7 inches diameter, or with an area of 1,001 inches, and a piston rod equal to one square inch in area, so that the effective area of the piston is equal to 1,000 square inches.

Now if one pound of ammonia is introduced below the piston at a temperature of 32°, or 493° absolute, and a vacuum is maintained above it, while the weight of 14,700 pounds (equal to the atmospheric pressure of 14.7 pounds on 1,000 square inches) is supported from it, then it will be found that the volume of the pound of gas at such pressure and temperature is equal to about twenty-one cubic feet, or 36,288 cubic inches, and that the piston will consequently stand 36.28 inches up from the bottom. Of course this is assuming an absolutely frictionless piston and piston rod for the purpose of the experiment.

If more heat is now allowed to pass into the gas until its temperature is doubled, that is raised to $493° \times 2 = 986°$ absolute, or 525° by the thermometer, and at the same time weights are gradually added in such a way as to maintain the piston continuously in the same (or No. 1) position, then it will be found that when the temperature has been doubled the pressure has doubled also, and that the weights that can be supported in such original, or No. 1, position will amount to 29,400 pounds. If, further, the temperature is trebled, then the piston in No. 1 position will support 44,100 pounds weight, and so on.

If the heat that is required to be communicated to the ammonia, to enable it to carry the double load, and to raise its temperature 493°, is measured, it will be found to amount to 192.8 thermal units, and if 192.8 is divided by 493, it gives .3911 unit, as the amount of heat which must be communicated to the gas for each degree rise of temperature. Therefore .3911 of a unit is said to be the specific heat of

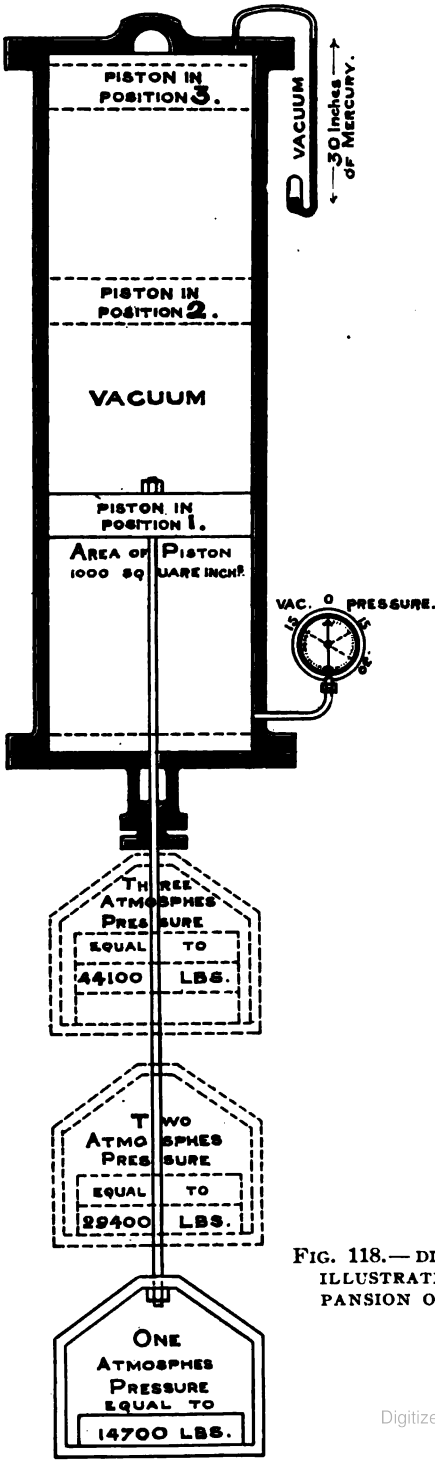


FIG. 118.— DIAGRAM TO ILLUSTRATE THE EXPANSION OF GASES.

ammoniacal gas at constant volume. Similarly a further accession of 192.8 thermal units is again required to be added, when 44,100 pounds is supported in the first position with the same volume of gas, but with its pressure trebled.

Let it be assumed that the cylinder is an absolute non-conductor, and that all this additional heat communicated is retained, the first impression of a student of the subject would be that the same amount of heat as is necessary to double the pressure of the gas and carry double the original weight, would raise the original weight alone to the second position; and further, that trebling the heat of the gas would treble its volume, and enable it to raise the 14,700 pounds to the third position. The vacuum, of course, being understood to be maintained above the piston throughout the whole experiment. Such is not the case however.

If now a second experiment is attempted, and when the piston is in the first position supporting 29,400 pounds, the additional weights are taken off (leaving only the original 14,700 pounds suspended), in the expectation that the gas at initial volume, and double its initial pressure and temperature, will expand to double its original volume and its initial pressure, it will be found that although the piston will certainly rise and lift the load as the weights are reduced to 14,700 pounds, it will stop a long way short of the double height indicated by the piston in position No. 2. To continue the experiment until the piston is raised to the double height it will be necessary to communicate additional heat to the gas, to the amount of $493 \times .1169$ unit, which $= 57.63$ thermal units. With such an amount of additional heat, the piston will raise the original weight the whole of the three feet to the second position, by doubling the volume of the gas beneath it. This additional heat, that is .1169 B. T. U. per pound of gas, is called the latent heat of expansion. When that amount of heat is added to the .3911 unit which represents specific heat at constant volume, it gives a total of .5080 thermal unit, or the specific heat of ammonia gas at constant pressure.

In carrying out the first part of the experiments it will be found that as the weights are taken off, the temperature of the gas will fall as the piston rises, although the cylinder

is non-conducting; and this fall of temperature represents a loss of thermal units exactly equivalent to the amount of mechanical work done in lifting the load, as it is reduced in weight.

The reason why more thermal units must be imparted to the gas in the second operation (although in both cases only one and the same pound of it is raised in sensible temperature, by exactly the same number of degrees) is easily comprehended in the light of the mechanical equivalent of heat; because in the second case there is external work performed in raising the 14,700 pounds of weight over three feet. It was from the consideration of these two different aspects of specific heat, in experimenting with gases, that Dr. Mayer is said to have first approximately deduced the value of the mechanical equivalent of heat, which was afterward more accurately determined by Joule, about the year 1842.

If we multiply 14,700 pounds by 3.0264 feet, the length of stroke of the piston, we obtain 44,488 foot-pounds as the amount of work done, and if this is divided by 493°, then $\frac{44488}{493}=90.3$. This number is the latent heat of expansion for ammonia, expressed in foot-pounds; and when it is added to 301.9, which is the specific heat of ammonia in foot-pounds at constant volume, it gives 392.2 as the specific heat in foot-pounds at constant pressure. This value, it will be recognized, is simply the specific heat in thermal units, viz., 0.5080, multiplied by the mechanical equivalent 772, any slight discrepancies in the fraction arising from the omission of small decimals.

To Mayer is due the credit that he arrived by abstract reasoning at results very close to those which Joule afterward confirmed by mechanical experiments. The mechanical equivalent of heat is now generally termed a "Joule," and designated by the letter J., as the equivalent of 772 foot-pounds, or 1 B. T. U.

The latent heat of expansion expressed in foot-pounds, for any gas, may very easily be found directly, when we know the volume of a given weight of the same at any temperature, or have the constants or co-efficients, such as is given in a table on page 181.

For instance, take a pound of air at 32° having a volume

of 12,387 cubic feet. It is evident that if such air was contained in a flexible bag, and the volume of the same was doubled by doubling the absolute temperature, then the whole weight of the atmosphere on one square foot would have to be lifted 12.387 feet. The atmospheric pressure of 14.7 pounds × 144 gives 2,116.8 pounds as the pressure per square foot; and this multiplied by 12.387 gives 26,220.8 foot-pounds, as the amount of work involved in the operation. Then 26,220 divided by 493° (which represents the number of degrees the temperature has to be raised) gives 53.18 as the latent heat of expansion for air, expressed in foot-pounds.

The specific heat of air expressed in foot-pounds is, therefore—

	Foot-pounds.
At constant pressure	183.45
At constant volume	130.3
Difference	53.15

The ratio of 183.45 to 130.3 is the same as that between .2777 and .1688, which is 1.408 to 1.

This ratio has been confirmed by experiments conducted by M. Masson, who liberated compressed air, and allowed it to expand back to its original temperature and deduced therefrom the ratio of 1 to 1.41, or 1 to $\sqrt{2}$.

This ratio is generally represented in the text books by the sign γ (Gamma, one of the letters of the Greek alphabet). The accepted value of γ for air is 1.401.

ON EXPANSION AND COMPRESSION.

In the work of a steam engine—expanding a saturated vapor, and in a compressor, such as the Linde machine—compressing what its makers term "humid" gas, any change of temperature which would be due either to alteration of volume of, or to the work performed by it, or upon it, is modified by the liberation or absorption of heat, that would not affect the operation with a perfect gas. In the steam engine, this arises from the setting free or liberation of heat from the entrained or suspended liquid, on the reduction of pressure causing re-evaporation during expansion, and in the compressor, by the absorption of the heat taken up to vapor-

ize the liquid held in the gas, which vaporization results from the increase of pressure and temperature.

In such cases, the compression and expansion appear more or less closely to follow Boyle's law, and in actual practice with the steam engine, it is generally considered to do so. Under that law, as has been shown, VP is a constant; the curve which represents the variation of the pressure throughout the stroke of the piston, is in such case a hyperbola, and the operation is termed "isothermal" compression or expansion.

To illustrate this graphically let it be assumed that we have gas at two atmospheres of pressure, or fifteen pounds

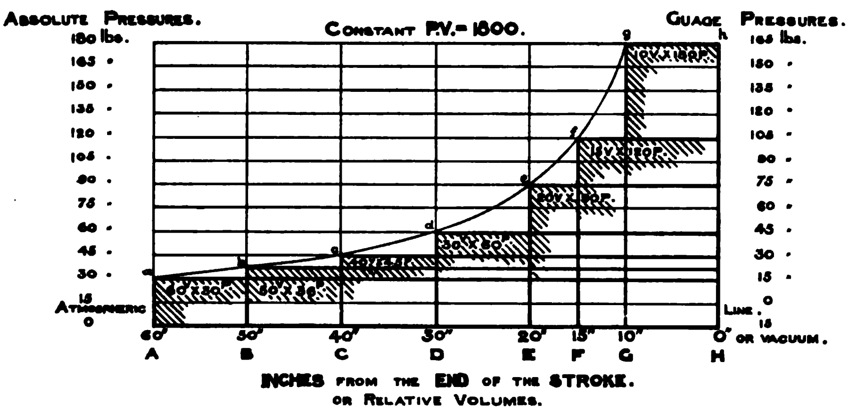


FIG. 119.—DIAGRAM OF ISOTHERMAL COMPRESSION AND EXPANSION OF A GAS.

by the gauge (the atmospheric pressure being taken at fifteen pounds, for the sake of round numbers), let the ratio of compression be 6 to 1, and instead of the cylinder being absolutely non-conducting as was assumed with Fig. 118, let it be a perfect conductor, and through external influences let the gas be maintained at a uniform temperature throughout the whole stroke of the piston.

Let the base line of Fig. 119 represent the zero of pressure or a vacuum, and its length, A H, sixty inches; this corresponding with the stroke of the piston, in a compressor or engine working with an expansion or compression of 6 : 1. It is of course apparent that if the diagram is to represent

expansion in an engine, that the stroke of the piston would be from right to left, the cylinder being filled at initial pressure for ten inches out of the full sixty inches before expansion begins, and then expanding through fifty inches to the end of the stroke.

As the purpose at present is to consider the action of a compressor, the journey must be made from the left to the right. With the piston commencing its stroke at a, the initial volume, or full cylinder, is represented by unit area of piston multiplied by sixty inches, or $V=60$. The initial pressure is thirty pounds, or $P=30$. Then $P\ V$ equals 1,800, which is graphically illustrated by the lower parallelogram, sixty inches long multiplied by thirty pounds high. When the piston has moved along the one-sixth of its stroke the volume of the cylinder will be reduced from sixty to fifty, and $\frac{1800}{50}$ gives thirty-six as the value of P for such volume. At forty inches, P becomes forty-five; at thirty inches, or half stroke, P has doubled its original value, and becomes sixty. Similarly, at e and f, P rises to ninety and 120, respectively; while at ten inches from the end, when the gas is confined to one-sixth of its original volume, P has risen to 180 pounds, or six times its initial pressure.

It is evident that the several parallelograms representing P V in all these different positions of the piston, are all of equal area; and as this corresponds with the construction of the hyperbola, a line which joins the points a b g will be a hyperbolic curve.

When a diagram of this character has been obtained directly from a cylinder by means of an indicator, the line A H is usually divided into a number of equal parts, say ten or more, by a set of parallel dividing rulers, and ten ordinates or heights are taken in the centers of each of those divisions. The mean height or mean pressure may then be ascertained by adding these ten values together, and dividing them by ten. When, however, there are no means of taking a diagram by an instrument, but the point of cut-off, and the initial and terminal pressures are known, then the mean pressure may be ascertained (without requiring special mathematical knowledge, or the construction of a diagram) by the use of hyperbolic logarithms as given in the table on opposite page.

Let R represent the ratio of compression and expansion.

H the hyperbolic logarithm of R.

P the mean pressure.

C the initial pressure before compression.

E the initial pressure before expansion.

Then for compression $P = C \times (1 + H)$

For expansion $P = \dfrac{E}{R} \times (1 + H)$

The following table gives hyperbolic logarithms for a number of different ratios of compression, but it must be understood that they only apply to the compression of any gas under the special circumstances of uniform temperature throughout the stroke with no allowance for clearance.

HYPERBOLIC LOGARITHMS FOR CALCULATING EXPANSION AND COMPRESSION OF GASES.

Portion of the stroke during which no expansion of the gas takes place.	Ratio of compression.	Hyperbolic logarithm.	Portion of the stroke during which no expansion of the gas takes place.	Ratio of compression.	Hyperbolic logarithm.
$\frac{9}{10}$	1.11	.104	$\frac{3}{10}$	3.33	1.203
$\frac{7}{8}$	1.14	.131	$\frac{1}{4}$	4.0	1.386
$\frac{8}{10}$	1.25	.223	$\frac{2}{10}$	5.0	1.609
$\frac{3}{4}$	1.33	.285	$\frac{1}{6}$	6.0	1.7917
$\frac{7}{10}$	1.42	.351	$\frac{1}{7}$	7.0	1.9459
$\frac{5}{8}$	1.6	.470	$\frac{1}{8}$	8.0	2.079
$\frac{6}{10}$	1.66	.507	$\frac{1}{9}$	9.0	2.1972
$\frac{1}{2}$	2.0	.693	$\frac{1}{10}$	10.0	2.302
$\frac{4}{10}$	2.5	.916	$\frac{1}{12}$	12.0	2.489
$\frac{3}{8}$	2.66	.978			

ADIABATIC COMPRESSION AND EXPANSION.

Reference has already been made to the effect which the humidity of the gas has in its effect on the operation of compressing ammonia, it being a special feature of some compression plants. It must not however be understood from this, that the refrigerating medium does more than approach to a perfect gas, without actually reaching that condition, even in those plants where the expansion coils of the refrig-

erator are of such ample surface (in proportion to the weight of ammonia to be evaporated) that the gas as supplied to the inlet of the compressor is technically "dry." In every-day practice, the line of the diagram, as taken by an indicator, from so called dry compressors, will not follow an exact adiabatic curve, because the walls of the cylinder must transmit some of the heat resulting from the compression, and this heat will be carried away by the jacket of water that nearly always surrounds the cylinders of dry compressors. Every unit of heat thus carried away, by reducing the pressure, of course reduces the amount of power necessary to work the compressor.

Notwithstanding this, if the engineer in charge of a compressor can set up the true adiabatic line, as well as the isothermal line of compression, upon the actual indicator cards which he takes from his cylinders, he will get then a better idea of the real work which his machine is doing, and also be able to judge whether improvements to it are either desirable or possible. A well fitted compressor should not only have indicator attachments and pressure gauges connected as closely as possible to the inlet and outlet branches, but should also have mercury wells for the insertion of thermometers close to the same connections. Direct readings of the gauges will give the initial and final pressures, P and $P + P^1$, from which R, the ratio of compression, can be deduced, and the relation of initial and final volume V and V^1. The thermometers will give the initial and final temperatures, which by the addition of 461° to each, gives T and $T + T^1$.

When instead of isothermal, it is adiabatic compression which takes place, then instead of $P\,V$ being constant, it is $(P \times V)\gamma$, or P multiplied by V raised to such power (Gamma) as is appropriate to the special gas under consideration, which is constant. These values are given in one of the columns of the table on the opposite page, and it is possible to prove, in accordance with the principles of logarithms, that the numeric ratio which the specific heat of a gas at constant pressure bears to the specific heat of such gas at constant volume (and which in the case of air is 1.408) corresponds with the index of the power to which $P \times V$

must be raised to give the true results of adiabatic compression, the equation being $P^1 = P\left(\frac{V}{V^1}\right)^\gamma$

In the case of ammonia, instead of the pressure under compression or expansion varying inversely as the volume,

PROPERTIES OF GASES USED FOR ARTIFICIAL REFRIGERATION.

	Gases. Temperature, 32° F. Pressure, 1 Atmosphere, or 14.7 Pounds Per Square Inch.		Sulphuric Ether.	Sulphurous Acid.	Carbonic Acid.	Air.	Ammonia.	Gaseous Steam.
1	Cubic feet in one pound		4.97	5.513	8.101	12.387	21.017	19.913
2	Pounds in one cubic foot		0.209	0.181	0.123	0.080	0.047	0.0502
3	Specific gravity, air being 1		2.586	2.247	1.529	1.000	0.589	0.622
4	Co-efficient (a) $VP = a\ (t + 461)$		0.1424 or $\frac{1}{7.019}$	0.1643 or $\frac{1}{6.089}$	0.2415 or $\frac{1}{4.1399}$	0.3693 or $\frac{1}{2.7074}$	0.6266 or $\frac{1}{1.59}$	0.5937 or $\frac{1}{1.684}$
5	Specific heat at constant pressure, K	In Thermal Units.	0.4810	0.1553	0.2164	0.2377	0.5080	0.4750
6	Specific heat at constant volume, S	In Thermal Units.	0.3411	0.1246	0.1714	0.1688	0.3911	0.3700
7	Latent heat of expansion or $K - S = L$	In Thermal Units.	0.1399	0.0307	0.0450	0.0689	0.1169	0.1050
8	Ratio of specific heats, $\frac{K}{S}$ or $\frac{k}{s}$ or γ		1.41	1.246	1.262	1.408	1.298	1.283
9	Specific heat at constant pressure, K	In Foot-pounds.	371.3	119.89	167.06	183.504	392.17	366.7
10	Specific heat at constant volume, S	In Foot-pounds.	263.3	96.19	132.32	130.31	301.92	285.64
11	Latent heat of expansion $K - s = L$	In Foot-pounds.	108.028	23.7004	34.74	53.19	90.25	81.06

it varies inversely as the volume raised to the 1.298 power. The following table gives the ratios of volumes and temperatures for air under twenty-five different grades of com-

pression, and has columns of differences, to enable intermediate grades to be dealt with—it is taken from a French work:

ADIABATIC COMPRESSION OR EXPANSION OF AIR.

Ratio of Greater to Less Pressures	Ratio of Greater to Less Absolute Temperatures.		Inverse of These Ratios. Ratio of Less to Greater Absolute Temperatures.		Ratio of Greater to Less Volumes.		Inverse of These Ratios. Ratio of Less to Greater Volumes.	
	Numbers.	Differ.	Numbers.	Differ.	Numbers.	Differ.	Numbers.	Differ.
1.2	1.054	48	.948	41	1.138	132	.879	91
1.4	1.102	44	.907	34	1.270	126	.788	73
1.6	1.146	40	.873	30	1.396	122	.716	57
1.8	1.186	36	.843	25	1.518	118	.659	48
2	1.222	35	.818	22	1.636	114	.611	40
2.2	1.257	32	.796	20	1.750	112	.571	34
2.4	1.289	30	.776	18	1.862	109	.537	30
2.6	1.319	29	.758	16	1.971	106	.507	26
2.8	1.348	27	.742	15	2.077	105	.481	23
3	1.375	26	.727	13	2.182	102	.458	20
3.2	1.401	25	.714	13	2.284	100	.438	19
3.4	1.436	24	.701	11	2.384	99	.419	16
3.6	1.450	23	.690	11	2.483	97	.403	15
3.8	1.473	22	.679	10	2.580	96	.388	14
4	1.495	21	.669	9	2.676	94	.374	13
4.2	1.516	20	.660	9	2.770	93	.361	12
4.4	1.537	20	.651	9	2.863	93	.349	11
4.6	1.559	19	.642	7	2.955	91	.338	10
4.8	1.576	19	.635	8	3.046	89	.328	9
5	1.595	86	.627	32	3.135	434	.319	39
6	1.691	77	.595	26	3.569	412	.280	29
7	1.758	70	.569	22	3.981	396	.251	23
8	1.828	63	.547	18	4.377	382	.228	18
9	1.891	59	.529	16	4.759	370	.210	15
10	1.950	..	.513	..	5.129	...	.195	..
1	2		3		4		5	

After what has been said it must be clear, that in the compression of any gas, the work which has to be done at every successive step or stage, to effect such compression, must add to the pressure, which would result from the simple reduction of volume under Boyle's law, by the addition of the heat units which are equivalent to such work. That being so, the next stage must start with a higher pressure than that which is simply due to $P\ V$ divided by V^1. The pressure at the end of each separate stage is dependent upon the work which is necessary to overcome the ever varying

pressure during such stage, and the equation in consequence involves the use of logarithms. It is however only necessary to make the steps or stages of the compression relatively small to be enabled to arrive at the adiabatic result, with a little more labor, by simple arithmetical calculation alone.

As the author is not aware that the method has ever been suggested before, an example may be given, in which some of the stages will be worked by a series of decreasing increments, and others by a system of trials; the proof of the result in all cases will be that the pressure arrived at is directly as the intrinsic energy in the gas, and inversely as its volume. So far as experiments have gone, the specific heat of gases is not seriously affected by difference of pressure and volume.

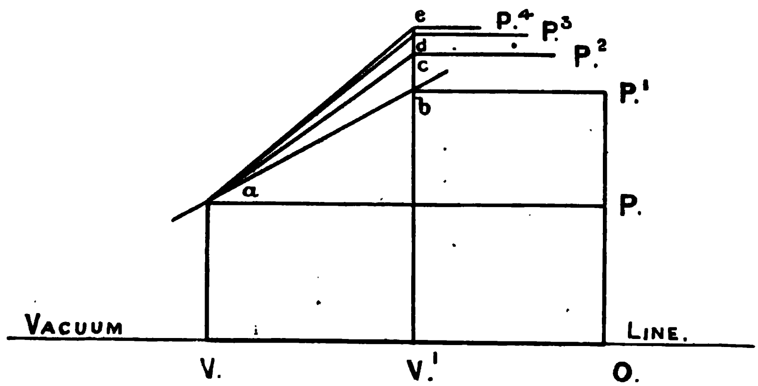


FIG. 120.—DIAGRAM ILLUSTRATING ACCESSION OF HEAT AND INCREASE OF PRESSURE BY COMPRESSION.

Let there be a cylinder of known area of piston, filled with unit weight of gas, of known temperature, pressure, and specific heat. Then the volume will be that which is due to the weight, at such temperature and pressure; and if contained in a full working cylinder, the volume divided by the area of the cylinder will give the length of stroke. The intrinsic energy of the contents (which may be called E) will be the product of the weight of the gas multiplied by the number of degrees of absolute temperature, and by its specific heat.

In Fig. 120, let the length of the horizontal line V O represent the initial volume of such weight of gas, and the

height P O its initial absolute pressure. Then when the volume is reduced to V^1 O, *without accession of heat*, the pressure will be increased to P^1 O, and the two parallelograms a P O V and b P^1 O V^1 will be of equal area. If the interval from V to V^1 is relatively small, the curve extending from a to b, and representing the increase of pressure during such compression, will approach so closely to a straight line that the mean pressure of the gas during its compression between the two volumes V and V^1 will practically be equal to—

$$\frac{P\,O + P^1\,O}{2}$$

If the mean pressure thus ascertained is multiplied by the area of the piston, it will give the mean resistance to it, or the mean force in pounds exerted by the piston of the machine during the operation. This force multiplied by the distance V to V^1, in feet (represented in the diagram by the area V a b V^1), will give the foot-pounds of work, or the amount of energy, exerted by the piston in effecting that stage of the compression. The number of foot-pounds thus arrived at, if divided by 772, will give the value of such energy in thermal units.

If the energy in the gas before compression, and with the piston at V, equals E, and the additional energy which is involved in compressing it from V to V^1 equals E^1, then the total energy in the gas at V^1 will be $E + E^1$ instead of E, and the temperature at b will be that due to $E + E^1$, and not that due to E. But if this is the case, and the pressure is directly as the temperature, the pressure at V^1 will not be that first assumed, and represented by the height of P^1, above O at the point b, but will necessarily be increased in the ratio of E to E^1, as the effect of the piston's work on the gas between V and V^1, and $\frac{P^1 \times (E + E^1)}{E}$ will give a value P^2 as the pressure due at such volume to the energy in the gas.

But if P^2 is the real pressure after compression, then the mean pressure during compression would be $\frac{P + P^2}{2}$ instead of $\frac{P + P^1}{2}$ which has just been assumed to be the case.

It is therefore necessary to proceed further, and ascertain the value of the work or energy, E^2, due to the area and stroke multiplied by the small difference of pressure represented by $\frac{P^2 - P^1}{2}$, and dealing with it in the same way as before (by adding to the already accumulated energy the additional energy represented by the triangle a b c), find a position P^3, from which the energy and consequent increase of pressure represented by the triangle a c d could be deduced and added to the gas. Before this is done, however, the quantities will have become so numerically small that P^3 will be found to coincide very closely with the value obtained by the use of logarithms. If still greater accuracy is desired, however, then, as the data are established for the area of the triangle a c d, the heat represented by the additional pressure P^3 may be added, and a fourth value P^4 be found, and so on to the infinitesimal.

From this it will be seen that the pressure due to adiabatic compression may be practically arrived at by a series of simple arithmetical additions; it may also be obtained by means of trials, in which the terminal pressure to each stage of compression is assumed. In the latter case, the heat or energy, necessary to do the work of compressing the gas to such assumed terminal pressure and temperature, is added to the initial energy; and the additional pressure due to such work, is then added to the increased pressure which is due simply to change of volume. If when this is done, the terminal pressure arrived at by the calculation corresponds with that which was assumed, it may be taken for granted that it is the correct one. If the result is higher or lower, then an indication will be given for further trial, which can be repeated until the result is sufficiently close for the purpose.

If the interval from V to V^1 in Fig. 120 is so relatively large that the line a b would have a sensible curve in it, then it is certain that the mean pressure during such stage of compression would be sensibly less than $\frac{P+P^1}{2}$, and therefore any results which might be obtained by considering it as straight, would be too high.

It may be said that such methods of calculation are useless, because a table of logarithms will give the same results in much less time than is required for the more lengthy and laborious calculation; but as a great many refrigerating engineers may not think so, and may prefer the simple to the more abstruse operation, it will perhaps be well to go further, and as an example apply these methods of calculation to a compression cylinder of a definite size, and a gas in every day use.

Let Fig. 121 represent a cylinder containing one pound of ammonia gas, at a temperature of 32° or 493° absolute, and a little over two atmospheres, or thirty pounds absolute

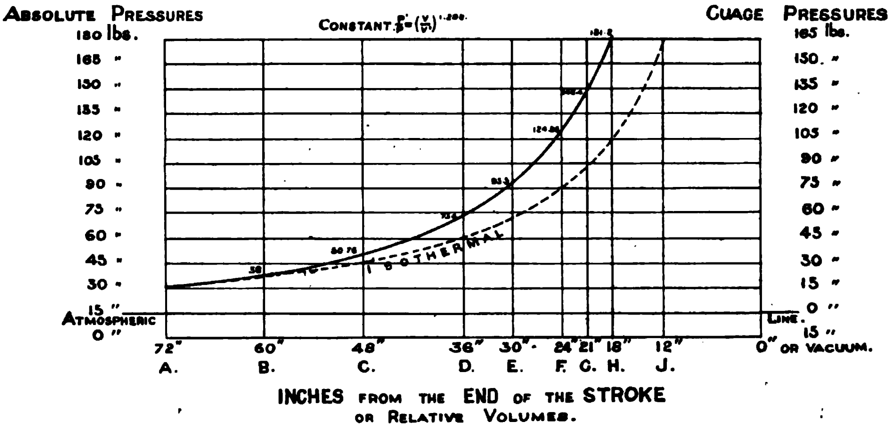


FIG. 121.—ADIABATIC COMPRESSION AND EXPANSION OF AMMONIA.

pressure; then according to table on page 191 the volume in cubic feet or V is equal to $\frac{T+461}{1.598\ P}$ The pressure P is thirty, and that multiplied by 1.598=47.94. Whence $\frac{493}{47.94}$ gives 10.29 as the volume of the gas in cubic feet, under the conditions stated.

If the working length of the cylinder, or the stroke, is six feet, then, $\frac{10.29}{6}$ gives 1.715 square feet for the cross section of cylinder, which, multiplied by 144, gives 246.9 (say 247) square inches, as the area of the piston.

The initial weight of the gas being one pound, the temperature 493°, and the specific heat .391, then the initial intrin-

sic energy of the gas must be $493 \times .391 = 192.76$ thermal units - - - - - - - - - - - (1)

If the compression of the gas from the full six feet length of the cylinder, be calculated through a series of stages, under one or other of the methods just suggested, and the results be compared with those obtained by the use of logarithms, it will possibly lead to a clearer comprehension of the specific heat of gases under different conditions.

In the diagram Fig. 121 there are seven stages of compression illustrated in the six feet stroke; viz., three of one foot each, two of six inches, and two of three inches each; until the gas is reduced to eighteen inches of the cylinder, or to one-quarter of its initial volume. It may be noted here, that with the accession of heat, a higher pressure is seen to be reached at four-fold compression, than is shown in Fig. 119, with six volumes compressed into one under constant temperature.

The total length of the cylinder in this case being six feet, the proportion occupied after the several stages of compression will be as follows:

Stage of Compression.	1	2	3	4	5	6	7
The length of the cylinder occupied by the gas............	5′	4′	3′	2′ 6″	2′	1′ 9″	1′ 6″
The ratio of original to new volume.....	1.2	1.5	2.	2.4	3.	3.42	4.
The ratio raised to the power γ, which for ammonia=1.298	1.267	1.692	2.458	3.116	4.162	4.948	6.04
The initial pressure of thirty pounds absolute being multiplied by these logarithmic ratios giving adiabatic pressure..............	38.01	50.76	73.74	93.48	124.86	148.44	181.2
The isothermal pressures being........	36.	45.	60.	72.	90.	102.6	120.

To calculate the adiabatic pressures for these same stages of compression in the absence of tables :—

Commencing with the initial pressure of thirty pounds, let it be considered that at the end of the first stage the pis-

ton has moved one foot, reducing the volume in the ratio of 6 : 5; and that the terminal pressure, by increasing in the ratio 5 : 6, would be thirty-six pounds from alteration of volume alone; then in such case the mean pressure on the piston during its movement would be about thirty-three pounds, because—

$$\frac{30+36}{2}=33.$$

This pressure, thirty-three pounds by 247 inches—the area of the piston--and by one foot stroke, gives 8,151 foot-pounds as the work of compression; which is equal to 10.43 heat units. Now, as the original energy in the gas (see 1) was 192.76 units, the accession of 10.43 more units would raise the energy of the mass to 203.18 units - - (2)

The pressure for constant volume is directly as the temperature or energy; and therefore the pressure of the gas, when the effect of this extra 10.43 units is taken account of, will not be thirty-six pounds as already arrived at, but—

$$\frac{36 \times 203.18}{192.76} = 37.9 \text{ pounds.}$$

But if the terminal pressure is 37.9 pounds it must upset the data on which the previous work was based, because if the terminal pressure is 37.9 instead of 36, then the mean pressure would be 33.95 instead of 33; and the amount of heat or thermal units which should be added for the work done, must be increased in like proportion—which is about 2.85 per cent. The 10.43 units when increased by 2.85 per cent amount to 10.72 units, and we can now start the calculation afresh, with 10.72 units as the measure of the additional heat due to the work of compression, the total energy in the gas being 192.76 + 10.72 = 203.48 units - (3)

$$\text{and } \frac{36 \times 203.48}{192.76} = 38 \text{ pounds pressure.}$$

This amount it will be seen is only one-tenth, or 0.1 of a pound, more than was taken as the basis of the second trial; and although the increment would be too small to have any practical value, still it is evident that by performing another operation to ascertain the increase of pressure that would be due to the additional temperature that would result from the additional mean pressure of .05 pound to the inch on the

compressor piston, the pressure of thirty-eight pounds would be actually increased by a small fraction. The result already attained, however, is so close (within .01, or the one-hundredth part of one pound pressure) to the pressure as calculated by logarithms, as to answer perfectly well for all practical purposes. In connection with the operation of a compressor, where ordinary pressure gauges and thermometers are used, the calculation of the pressure to several places of decimals would be useless, because such accuracy would be nullified by the conditions of actual work, and by the relative imperfections of the instruments employed.

Commencing the second stage with gas at thirty-eight pounds pressure, and an intrinsic energy of 203.48 units, the movement of the piston through the second foot would reduce the volume in the ratio of 5 : 4, and the pressure due to such reduction of volume alone would be $\frac{38 \times 5}{4} = 47.5$ pounds.

For this stage let it be assumed for the purpose of trial that the terminal pressure will be fifty pounds instead of 47.5, then as $\frac{38+50}{2} = 44$, the mean pressure must be taken as forty-four pounds, instead of 42.75 pounds to the inch.

A pressure of forty-four pounds on a piston area of 247 square inches, through twelve inches of space, gives 10,868 foot-pounds, $=14.07$ thermal units.

The total energy at the end of the former stage (3) was 203.48 T. U., and 17.07 added to this, gives 217.55 units total energy - - - - - - - - - (4)

The temperature being as the amount of energy, and the pressure as the temperature,

then $\frac{47.5 \times 217.55}{203.48} = 50.77$ pounds as the terminal pressure.

This differs by only one-tenth of 1 per cent from that calculated by the logarithm ratio, viz., 50.76 pounds.

For the third step, reducing the volume in the ratio of from 4 to 3, and initial pressure 50.76 lbs., $\frac{50.77 \times 4}{3} = 67.6$ pounds as the pressure due to reduction of volume alone.

For trial, as to the energy or work required for the actual compression, assume 72.25 pounds to be the ultimate or terminal pressure. Then $\frac{50.77+72.25}{2}=61.5$, which will be assumed as the mean pressure during the operation.

The area of piston in square inches, 247, multiplied by 61.5 pounds through one foot, gives 151,905 foot-pounds, or 19.6 thermal units, as the equivalent of the work of compression for this stage.

The energy in the gas at the end of the previous stage (4) was 217.55 units, and adding to this 19.6 additional units as above, gives a total energy of 237.15 thermal units - (5)

Then $\frac{66.6 \text{ lbs.} \times 237.15}{217.55}=73.6$ lbs. pressure for half stroke.

This is over the pressure assumed, and indicates that the assumption was too low; it is therefore slightly below the pressure found by means of logarithms, viz., 73.74 pounds.

If another trial is made and the pressure is assumed to be 73.50, instead of 72.25, then the result will come out practically correct.

Having so far compressed the gas to one-half of its original volume, or into three feet length of the cylinder, let the next foot be made by two stages of six inches each.

The initial pressure for the stage is 73.7 pounds.

The energy in the gas, from (5), is 237.15 thermal units.

The compression from three feet to two feet six inches is in the ratio of 6 : 5. Then $\frac{73.7\times 6}{5}=88.44$ pounds as the pressure due to change of volume only.

The mean pressure on the piston for such an increase would be $\frac{88.4+73.7}{2}=81.07$ pounds.

As the stroke is only six inches, with area of piston as before, $\frac{81.07 \text{ pounds} \times 247 \text{ inches area}}{2} = 10{,}012.14$. That is 10,012.14 foot-pounds, is equal to 12.97 thermal units.

The initial energy of the gas for the stage was 237.15 units as - - - - - - - - - - (5)

Therefore the terminal energy is $237.15 + 12.97 = 250.12$ - - - - - - - - - - - (6)

$\frac{88.44 \text{ lbs.} \times 250.12}{237.15}$ gives 93.27 lbs. terminal pressure.

It is here evident that much too low a figure was assumed for the terminal pressure in taking 88.44 pounds, because the calculated pressure so far has reached 93.27 pounds, and therefore the amount of energy added to the gas as equivalent to the work done on it was too small in at least the same proportion. The actual increase in the work done may be approached closer by a sum in proportion, and—

$\frac{12.97 \text{ units} \times 93.27}{88.44}$ gives 13.67 units as more nearly the equivalent of the work done than 12.97 units.

Trying again with this additional energy allowed for, $237.15 + 13.67 = 250.82$ units total energy - - (7)

Then $\frac{88.44 \times 250.8}{237.1}$ gives 93.5 pounds as terminal pressure.

The logarithm pressure is 93.42 pounds, and the slight excess probably arises from the mean pressure being somewhat less than an arithmetical mean between the initial and terminal pressures.

Commencing the second half of the fourth foot of the piston's stroke with a pressure of 93.5 pounds, the volume will be reduced from two feet six inches to two feet, or in the ratio of 5 : 4; and the pressure for change of volume will be raised proportionately.

$\frac{93.5 \text{ lbs.} \times 5}{4} = 116.87$ lbs. pressure for change of volume alone.

$\frac{116.87 + 93.5}{2} = 105.18$ arithmetical mean pressure during compression.

The stroke being six inches only—

$$\frac{105.18 \times 247}{2} = 12{,}989.7 \text{ foot-pounds}, = 16.82 \text{ units.}$$

The heat energy before compression was 250.82 units and $250.82 + 16.82 = 267.64$, total units - - - - - (8)

$$\frac{116.87 \text{ lbs.} \times 267.64}{250.82} = 124.6 \text{ lbs.}$$

But the pressure on which the accession of heat was based was only 116.87, instead of 124.6 pounds. We have

still therefore to allow for 7.73 pounds pressure, and consequently the accession of heat due to compression instead of being 16.82 units will approximate closely to—

$$\frac{16.82 \times 124.6}{116.87} = 17.93 \text{ units.}$$

Commencing again, 250.82+17.93=268.75 units - (9)
as the total energy in the gas at the end of the stage.

$$\frac{116.8 \times 268.75}{250.82} = 125.01 \text{ lbs. pressure.}$$

The logarithm calculation gives 124.86, showing that the mean was taken a little too high.

As the curve in the figure becomes more pronounced at the high ratios of compression, greater accuracy will be secured by taking two intervals of three inches each, when reducing the intervals from two feet to one foot six inches. First, taking from two feet to one foot nine inches, the ratio is as 8 : 7.

$$\frac{125 \times 8}{7} = 142.85 \text{ lbs., due to change of volume alone.}$$

Assuming an ultimate pressure of 148 pounds, take the mean pressure at 136.5 pounds, then the stroke being the fourth part of a foot—

$$\frac{247 \times 136.5}{4} = 8{,}428.8 \text{ foot-pounds,} = 10.91 \text{ units.}$$

The energy of the gas at twenty-four inches was 268.75 units - - - - - - - - - - - (9)
and 268.75+10.91=279.66 - - - - - - (10)

$$\text{Then } \frac{142.85 \text{ lbs.} \times 279.66}{268.75.} = 148.6 \text{ pounds.}$$

The result as given by logarithms =148.4 pounds.

Take next step, from one foot nine inches to one foot six inches of cylinder, the ratio being 7 : 6,—

$$\frac{148.5 \times 7}{6} = 173.25 \text{ pressure due to volume alone.}$$

$$\frac{173.25 + 148.5}{2} = 160.8 \text{ mean pressure.}$$

$$\frac{160.8 \times 247}{4} = 9{,}929.4 \text{ foot-pounds} = 12.84 \text{ units.}$$

The heat energy before compression was 279.66 - (10)

Then $279.66+12.84=292.50$ total units - - (11)

$$\text{And } \frac{173.25 \times 292.5}{279.66}=181.2 \text{ lbs. to the inch.}$$

The result as given by logarithms $=181.2$.

If the condenser pressure is taken for the terminal pressure in any compressor, and it is required to ascertain the volume of the gas as expelled, or the point where expulsion commences, it can be found, by working up step-by-step from the back pressure and temperatures, until it is reached. If the elementary methods thus far explained, for the benefit of weak mathematicians like the writer, should give the reader a taste for deeper research into the subject, there are plenty of advanced works on thermodynamics now available for him to wade into. It is not generally found, however, that deep academic research, and great practical skill and experience in the operation of machinery, go hand in hand; the author at any rate has never yet met them combined in the one engineer.

Life is too short, and the world is too full of trouble, for a single individual to be able to know everything even about the machinery of refrigeration, although, perchance, you may occasionally meet a man who thinks he fills the order.

CHAPTER XVII.

STEAM BOILERS FOR COLD STORAGE AND ICE MAKING.

Except in the small minority of cases where ample water power is at hand, artificial refrigeration, through the instrumentality of a compressor, is absolutely dependent upon the boiler as the mainspring of its operations. Its efficiency and economy become therefore of vital importance in such connection. The subjects connected with boilers are however so varied and extensive, and they have already such a considerable literature of their own—wherein design, construction, use and maintenance are fully dealt with—that it may be considered not only rash but futile to attempt to compress any useful information connected with them into the compass of a single chapter. On the other hand it is possible that a few things may be said which, without going too fully into details, are pertinent to the interests of those who are connected with refrigeration and ice making machinery.

Like every other steam user the owner of a compressor is sure to be full of cares in connection with his machinery, and when it comes to the boiler, there are several points about which he may fairly be anxious. *First*, That his boiler should be economical in initial cost. *Secondly*, That he shall obtain from it as many pounds of steam as possible, for every pound of coal he pays for. And *Thirdly*, That it should cost the minimum amount for attention, maintenance and repairs.

Sometimes it is desirable—as with other industries—that the boiler of an ice factory should occupy as little space as possible, and be independent of brick setting. In other cases, as when the boiler has to be set up in the close neigh-

borhood of refrigerating tanks or cold chambers, the heat radiated from the boiler and its setting is not only a direct loss of power, but an indirect loss also, because by heating the surroundings, it increases the work of the compressor, which has to pump such heat out again.

THE WATER TUBE BOILER.

When we come to investigate the relative cost of different types of boilers, we first note that the evaporation of water into steam is very largely a question of having such water contained in a vessel, and in contact with one side of a metal plate, which plate has its other side exposed to direct radiation from the combustion of fuel, or to the heated gases resulting therefrom. It then becomes evident, as metal is generally sold by the pound, that *prima facie*, the thinnest boiler will be the cheapest. A ton weight of metallic water-vessels, in the form of small tubes, will certainly afford two or three times the heating surface that a ton of plates in an ordinary big boiler shell will do, and therefore, other things being equal, water tube boilers should be the cheapest form to construct for any given power. There is no doubt, moreover, as to their possession of other good qualities, although such are often exaggerated by persons interested in the sale of them. On its average merits, however, the water tube boiler has undoubtedly come to stay.

The conditions are not so favorable to water tubes when the steam user has a supply of water impregnated with minerals, which lines them up with a casing or coating almost like marble. Such scale seriously obstructs the conduction of heat, so that the coal bill may easily be doubled, or the owner may have to pay more for keeping the boiler tubes clean than the interest on the boiler itself comes to, if it is not done in a proper and scientific manner.

There are now, however, many special appliances provided for boring out the deposit in water tubes, some of which, operated by tube cleaner companies, such as the Union Boiler Tube Cleaner Co., of Pittsburg, Pa., U. S. A., are so efficient that the removal of the scale becomes a comparatively simple affair. The necessity for this cleaning is forcibly shown by certificates that boilers after being cleaned had risen from 24.8 per cent to 100 per cent evaporative

efficiency; or, in other words, had by fouling lost 75.2 per cent, which was restored by the application of a cleaner for a few minutes to each tube.

If the water is of an unimpeachable character, or if the deposit can be thrown down in separate vessels—either by heating it in an exhaust steam feed heater, as Figs. 144 and 145, or one to heat it to full boiler temperature—before feeding it into the boiler itself, as Figs. 146 and 147, then the water tube boiler should give satisfaction. They are specially adapted for high pressure, look well from the outside, and

FIG. 122.—WATER TUBE WITH SCALE INSIDE.

FIG. 123.—FIRE TUBE WITH SCALE OUTSIDE.

will evaporate as much water as any other well proportioned boiler as long as they are kept clean; but, unfortunately with no other type are there such difficulties in the way of speedily removing the hard scale, or deposit, which rapidly lowers the evaporative efficiency.

As this may be thought a strong thing to say, Figs. 122 and 123 should be studied; they show that the deposit in the water tube is absolutely bound in like an arch, and cannot be moved until the key is forcibly broken.

The scale however on the outside of the fire tube will crack and drop off with a tap of a hammer, or fly by the sudden expansion of the tube, when a red hot heater is passed through it. With ordinary hand appliances and hard scale it is no uncommon thing for three men to be half an hour cleaning one water tube; and this costly process had considerable weight in restricting the use of water tube boilers in many localities before the resources of inventors provided

means for the simple and effective removal of the deposit.

So many fine water tube boilers are now made that it would be invidious to mention any by name. Purchasers should look for rapid circulation over the heating surfaces, and quiet water in the mud drums, as well as simple arrangements for cleaning and removing the tubes.

MULTITUBULAR BOILER.

The most formidable all-round rival to the water tube class, seems to be the ordinary underfired multitubular boiler, which appears to hold the premier place in the ice factory, not only in the United States, but in Australia also. When this boiler is properly proportioned and properly set, and is supplied with good water, it is, in the author's opinion, the best, the cheapest and the simplest, for its efficiency, of any boiler made. If such a boiler is worked with liquid mud, instead of water, then it should cause no surprise to see carbuncles form on the shell over fire; such things *have* happened through carelessness or ignorance, and will probably occur again. Further it will not do to blow these boilers off half an hour after shutting down at week's-end on Saturday, and then fill them up again the same afternoon, when part of the bottom has become red hot from the incandescent furnace walls. A new $4,000 boiler was thus ruined in one act, by a fireman's inexperience, to the author's personal knowledge. Again these boilers—often as much as five-eighths of an inch thick over the fire—are very susceptible to the action of cylinder oil, when it is returned with the feed water. The oil is apt to form a leathery skin, which keeps the water from direct contact with the plates, and is highly non-conducting. One battery of water works boilers in Australia, at any rate, are known to have become burnt in their bottoms through this cause. Perfect filtration and separation of the oil is absolutely necessary for the underfired boilers if the condensed water is returned as feed. Mr. Blechynden, who made exhaustive experiments on the transmission of heat through plates, has shown that the slightest deposit of grease or dirt on the plates causes a large fall off in the transmission of heat through them.

As made in the United States, and illustrated in the catalogues of refrigeration and other engineers, multitubular

boilers for a given horse power, seem to be smaller in diameter and to be crammed much more closely with tubes, than is customary in Australia.

The pages of *Ice and Refrigeration* show, that at least one boiler explosion at an ice factory has been attributable to this close packing of tubes, which caused a bad circulation, and a jamming of dirt, in the narrow space between the tubes and the shell. It seems to be often forgotten that the length and diameter of the tubes in a boiler should be determined by the pressure of the draft the chimney can produce. If the draft is light, and the tubes are small and long, what wonder they soon foul up and require constant brushing or steam blasting? Australian boiler builders appear generally to favor larger tubes than either English or American makers, four inches diameter being a common size, and it is usual to set them with space down the middle of the boiler wide enough for a lad to get down. This allows for easy scaling, and assists circulation. It is an undoubted fact that numbers of these boilers, which were originally stuck as full of tubes as they could hold, have had their evaporative efficiency improved by taking out a row or two in the wings, or down the center of the barrel.

The suspension of multitubular boilers over their furnaces has formed the theme of several engineering papers, and in the discussions thereon great differences of opinion as to small details have been manifested; but there is a general consensus of opinion, that such boilers should be suspended from above, and not rest on the side walls of the furnace. The long projecting brackets sometimes attached to the shell, to rest on the top of the side wall, must throw a great wrenching strain on the plates, and an angle iron extending the whole length of the boiler is preferable for this purpose if the boiler is not to be hung from girders overhead.

In the mounting and setting of multitubular boilers, there is room for great differences of opinion; every maker more or less following his own ideas.

For country use in Australia, and in sizes up to about twenty nominal horse power, these boilers are made with a sheet iron casing lined with fire brick, and are known as colo-

nial boilers. They are very handy and portable, as will be seen by Fig. 124, but as they waste fuel by diffusing the heat of the furnace around them through the four and one-half inches thick of fire brick walls, they can hardly be recom-

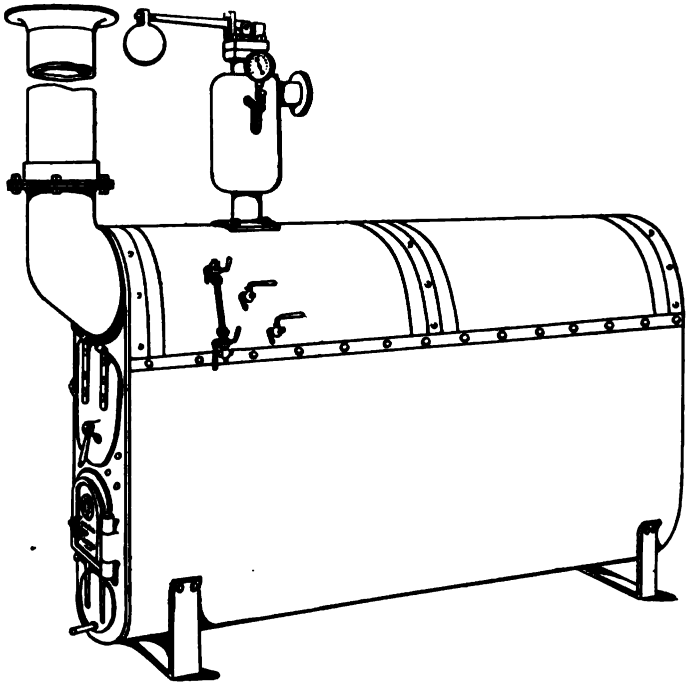


FIG. 124.—UNDER-FIRED BOILER—COLONIAL TYPE.

mended in connection with refrigeration, except for very small plants.

SPECIAL MULTITUBULAR BOILER FOR ICE FACTORY.

As a direct contrast to the colonial boiler just referred to, Figs. 125, 126, 127 and 128 show four views of a multitubular boiler designed by the author, with a brick work setting specially suited for refrigerating houses. A number of these are working in Sydney, and are giving great satisfaction, although not in connection with refrigeration.

As will be seen, the air for combustion is taken through the hollow side walls to the ash pit; and all the radiant heat thus intercepted is returned in the heated air supplied to the furnace. This is of course a double advantage, because

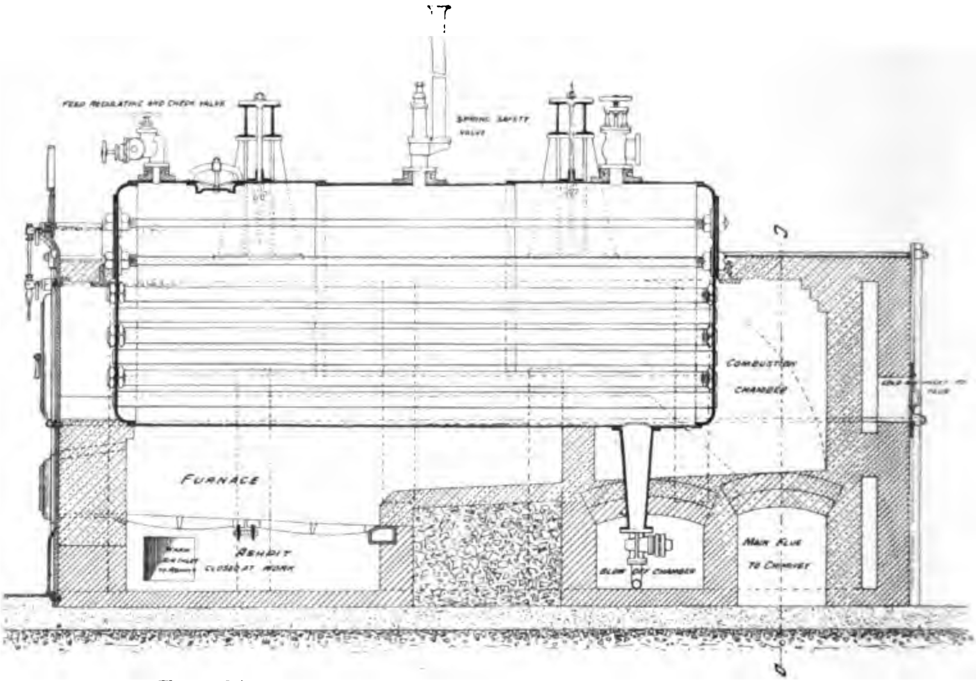


FIG. 125—MULTITUBULAR BOILER—LONGITUDINAL SECTION.

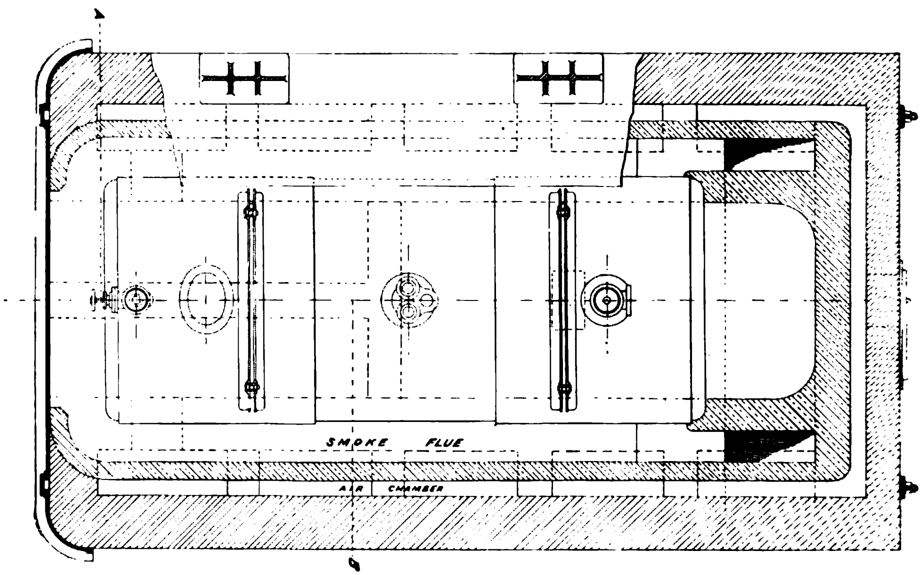


FIG. 126—MULTITUBULAR BOILER—PLAN.

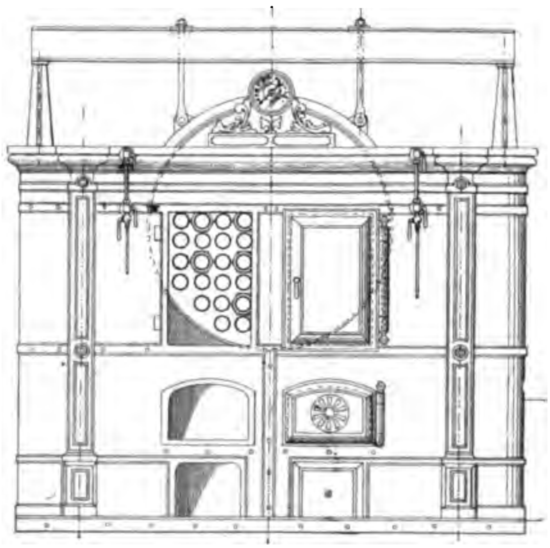


FIG. 127—MULTITUBULAR BOILER—DOUBLE FLUE SETTING —FRONT ELEVATION.

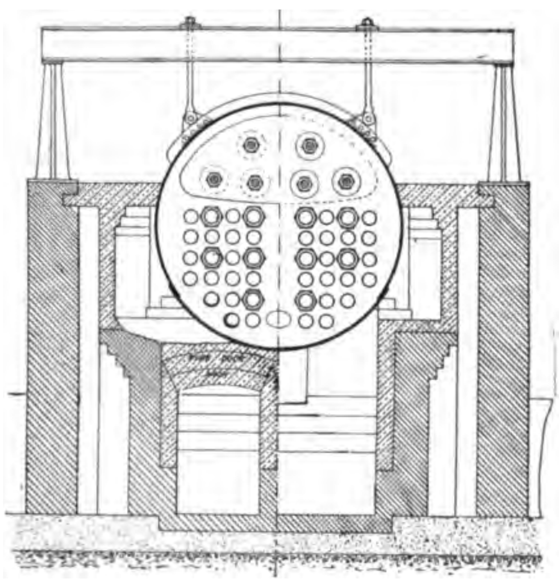


FIG. 128—MULTITUBULAR BOILER—DOUBLE FLUE SETTING—SECTIONAL VIEW.

there is first, better combustion from the heated air delivered to the fuel; and secondly, by the interception of the radiant heat, the walls on the outside of the brick setting are kept cool. A third advantage is, that the boiler can be shut down from six o'clock in the evening to six o'clock next morning without losing more than a few pounds of steam.

If Figs. 125 and 127 are examined, it will be seen that the ash pit doors have no hinges; but have a planed groove at bottom, which slides on a V-shaped rail. It is very hard to understand why so many boilers should be made with their ash doors hinged, so that when they stand open there is a direct inducement for the fireman to break his shins over them. Apart from the slovenly appearance, when they open at all angles on the floor plates, it is really much easier to regulate the draft when such doors slide than it is when they are hinged. In this particular setting the doors are always kept closed (except when cleaning out the ashes), because the air for combustion enters the regenerative casing by the regulator at the back, and leaves for the ash pit by the openings under the fire bars—see Figs. 125 and 126. With these underfired boilers, plenty of space should be left at the rear for a large combustion chamber, to permit the thorough admixture of the gases; otherwise many of the tubes may have a defective supply of oxygen, and fire will show at the front end when the tube doors are opened, or even at the top of the chimney—a sure sign of something very wrong.

THE CORNISH BOILER.

Human ingenuity has been at work for nearly a century designing new patterns of steam boilers. Their number is now so great as to pass any one man's knowledge, and their complicated construction, any one man's power of understanding. For all that, the most fearful and wonderful designs are still being continually evolved from inventors' brains. Some of these get so far as to be made and tested, while a few reach the advertising pages of the engineering journals.

The very best advice that can be offered to any steam user, who is not himself an expert, is to have nothing to do with any revolutionary invention; simplicity is the great

desideratum in a boiler, and complication should be shunned. There appears after all these years to be only one man who is entitled to immortality in connection with this branch of engineering, and that is the father of the high pressure steam boiler, and of the locomotive—Richard Trevithick.

One hundred years ago, in 1799-1800, this great Cornishman was bringing his high pressure "puffing" engines into competition with Boulton & Watt's condensers. The "hearse" or "wagon" boiler of his rivals had superseded Newcomen's "pot" boilers; but however good the wagon boilers might be for one or two pounds pressure of steam, they were utterly useless for the twenty-five or thirty pounds which the "puffers" worked at. This led Trevithick to introduce the horizontal cylindrical boiler, with a tubular furnace and flue, which is now, after a whole century of use, absolutely the same as Trevithick left it so far as form is concerned; and it is still known as the Cornish boiler. These pages are hardly the place in which to pay a tribute to this great inventor of engines, boilers, pumps, steam whims, etc., and also of the locomotive, which anticipated Stephenson by many years. Fortune favored Watt and Stephenson however, and public opinion has almost made gods of them, while Trevithick's fame seems fair to be forgotten. Neither Watt nor Stephenson appears to have had the mechanical genius of Trevithick, and it is doubtful if the world's real debt to the two together, is as great as it is to the rugged Cornishman. Trevithick however did not possess that faculty of *generalship*, which is at the present day—just as it was in his own time—a greater factor than either genius or mechanical skill in securing honors and pecuniary rewards.

Trevithick's Cornish boiler is still as good a one as can be obtained for the work of refrigeration, where there is plenty of ground space, and where first cost is not so important as ultimate economy. Such boilers are made by thousands every year, and are used all over the world, as they will run for lengthened periods with less attention than some of the modern patent boilers require every week. In the best boiler builder's work, all angle irons are dispensed with, and the boiler ends are deeply flanged from steel plate circles. The furnaces are all welded up in lengths, without rivets or

longitudinal seams, and should be made with the Adamson joint, or be corrugated after one of the patents shown by

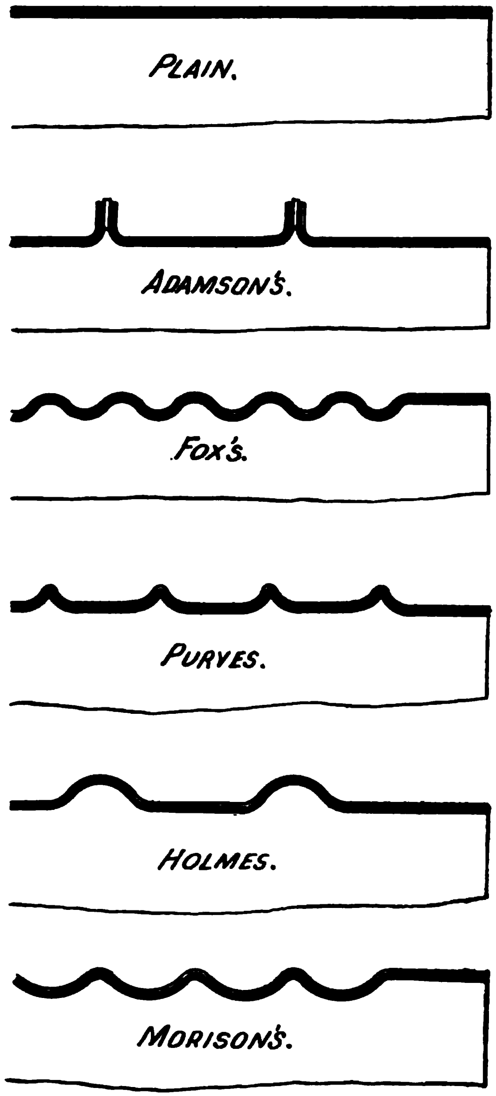


FIG. 129—VARIOUS PATENTED SYSTEMS FOR STRENGTHENING BOILER FLUES TO RESIST COLLAPSING PRESSURE.

Fig. 129. The flue behind the furnace should be made the same way, and is further strengthened generally by the

insertion of water tubes. Fig. 130 shows the front of a modern Cornish boiler fitted with an automatic stoker.

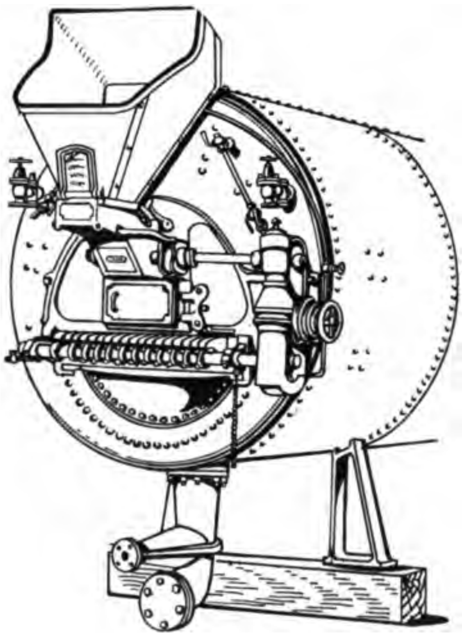


FIG. 130.—CORNISH BOILER WITH AUTOMATIC STOKER.

The following table gives the weight and evaporation efficiency of three sizes of modern Cornish boilers by English makers:

Diameter of shell.	Diameter of flue.	Length.	Evaporation per hour.	For 160 Lbs. Pressure.		For 140 Lbs. Pressure.	
				Weight. Boiler.	Weight. Fittings.	Weight. Boiler.	Weight. Fittings.
5′ 6″	3′ 0″	16′ 5″	2,225 lbs.	14,000 lbs.	6,720 lbs.	17,248 lbs.	6,950 lbs.
5′ 6″	3′ 0″	20′ 6″	2,950 "	16,352 "	7,050 "	20,160 "	7,286 "
6′ 3″	3′ 6″	20′ 6″	3,700 "	22,960 "	8,170 "	26,880 "	8,406 "

If sixteen pounds consumption of steam per horse power per hour is allowed for the 140 pounds pressure boilers, then their horse power comes out at 139, 184, and 231. Allowing fourteen pounds consumption for the 160 pounds pressure, it makes the horse powers 159, 210, and 264, respectively.

THE LANCASHIRE BOILER.

When Trevithick boilers are made six feet or more in diameter, they are generally fitted with two furnaces instead of one, and are then called Lancashire boilers. Fig. 131 is a longitudinal section of a modern Lancashire boiler, suitable for 140 pounds pressure to the square inch, fitted with Galloway tubes in the flues. It is a common thing to hear any ordinary water tubes, which cross a horizontal or vertical furnace, called "Galloways"; the essence of the Galloway tube however, is its conical form, so made in order that the flange at the small end may go through the hole cut for the large end. The small flange is thus fitted inside the flue, as seen in the section, while the large flange fits on the outside. The Galloway company of Manchester, England, are among the most celebrated makers of land boilers in the world, and their special "Galloway boiler" is a modification of the Lancashire form; in this system the two furnaces merge into a single kidney-shaped tube or flue, which is filled with taper water tubes, vertical and inclined.

The following table gives particulars of eight sizes of Lancashire boilers (two flues) for 105 pounds working pressure. From eighteen to twenty-six pounds of coal may be effectively burnt on each square foot of grate per hour, with a good chimney draft:

Diam. of boiler.		Length of boiler.	Diam. of flues.		Length of grates.		Grate surface.	Effective heating surface.	Approximate weight of boiler and mountings for 105 lbs. working pressure.		
Ft.	In.	Ft.	Ft.	In.	Ft.	In.	Sq. Ft.	Sq. Ft.	Tons.	Cwt.	Pounds.
6	6	18	2	6	4	6	25.5	420	11	9	25,648
6	6	27	2	6	6	0	30	633	15	0	33,600
7	0	21	2	9	5	0	27.5	541	13	12	30,464
7	0	30	2	9	6	0	33	775	18	2	40,544
7	6	21	3	0	5	0	30	585	15	12	34,944
7	6	30	3	0	6	0	36	839	20	11	46,032
8	0	21	3	3	5	0	32.5	626	17	7	38,864
8	0	30	3	3	6	0	39	898	22	18	51,296

Fig. 132 shows a front view of Fig. 131. The right hand furnace front being removed, allows the crossed Galloway tubes to be seen, and Fig. 133 is a section of the Galloway

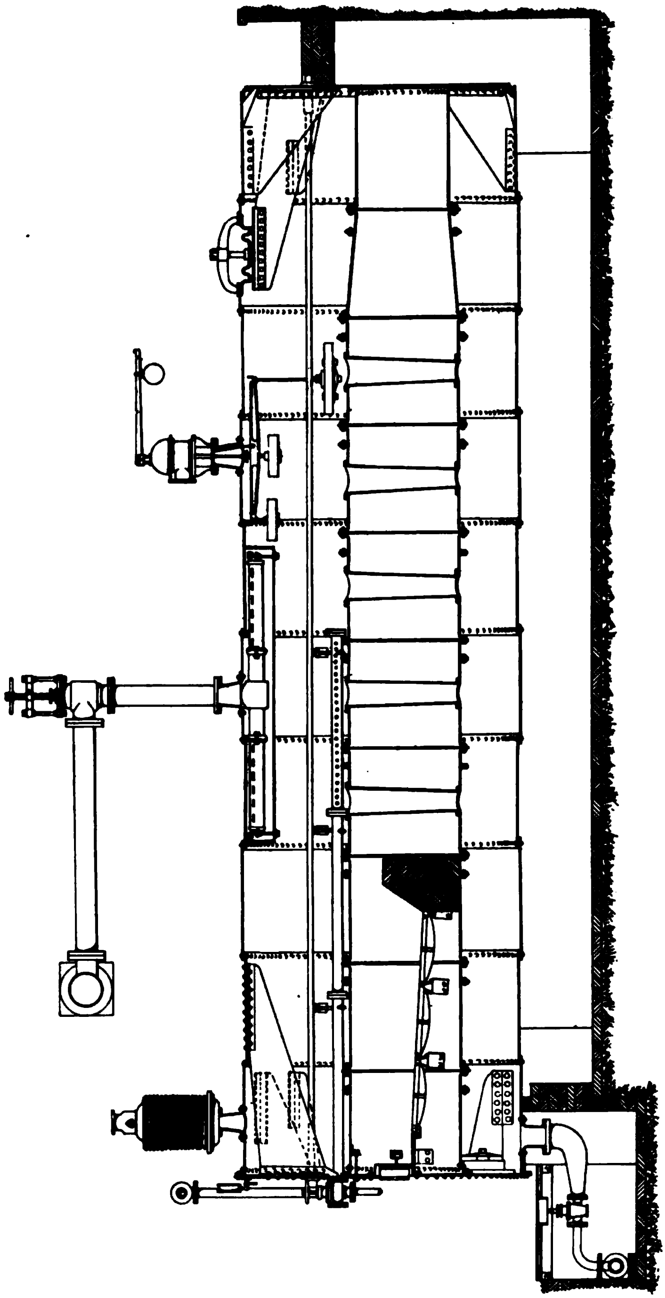


FIG. 131.—LONGITUDINAL SECTION OF LANCASHIRE BOILER WITH GALLOWAY TUBES.

patent boiler, with two furnaces uniting in one wide flue, filled with their special water tubes. It will be noted that in Fig. 131 there is a perforated feed pipe, an anti-primer for

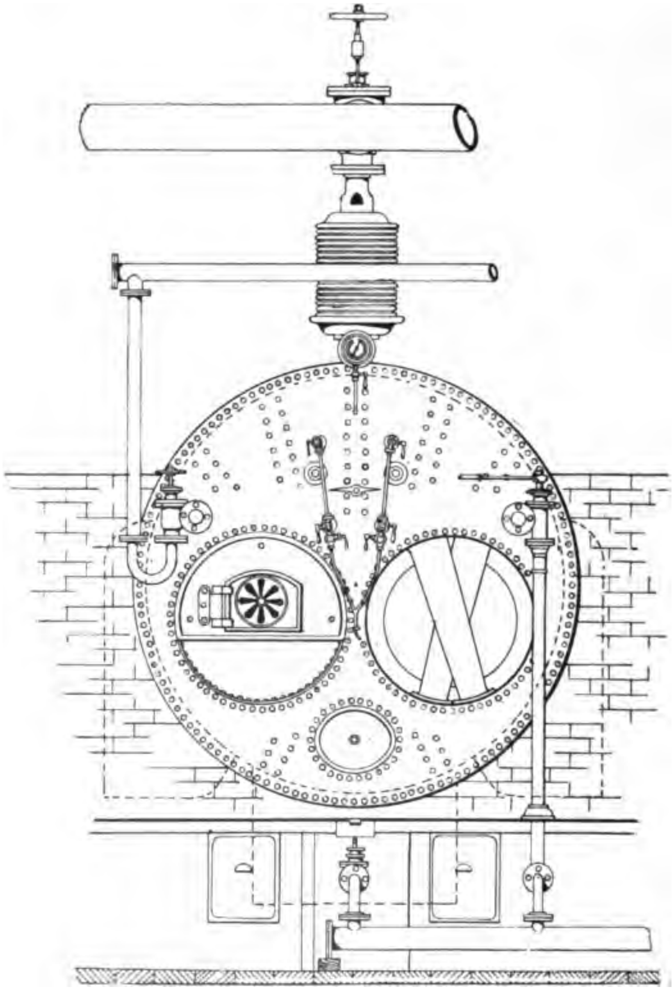


FIG. 132.—LANCASHIRE BOILER—FRONT ELEVATION.

taking dry steam, a high and low water alarm, dead weight safety valves, and a corrugated man hole door, all important.

CORNISH TUBULAR BOILER.

A modification of the Cornish boiler, which is daily growing in favor, is shown in the four illustrations Figs 134 to 137,

which represent a boiler specially designed by the author, for using water which makes a very hard deposit. There are several wide departures in it from common practice, the principal of which is the placing of the furnace to one side, instead of in the center of the shell.

This arrangement gives great facilities for the examination and cleaning of the inside, and also promotes better circulation. The multitubular arrangement of the back breaks up the gases, and by the increase of heating surface enables the whole boiler to be materially shortened. Two

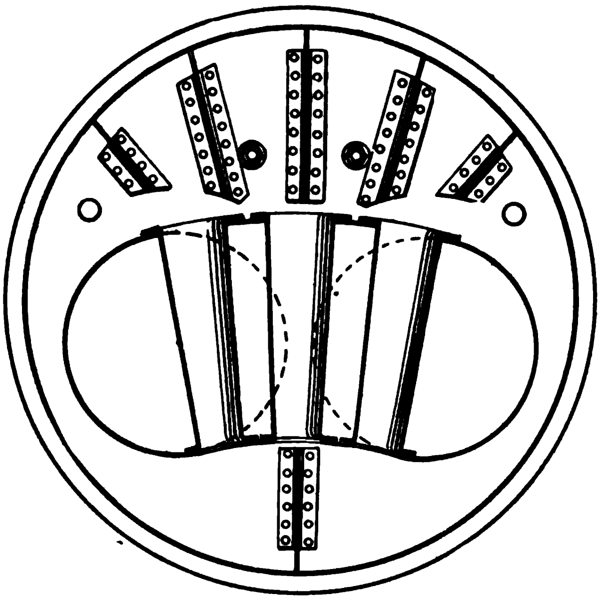


FIG. 133.—SECTION OF GALLOWAY BOILER.

eminent English authorities have certified that a boiler of this design in use at the office of a London daily paper had an efficiency equal to the evaporation of 10.15 pounds of water from 212° per pound of coal consumed.

By the adoption of one large four-feet furnace instead of having two furnaces each two feet six inches diameter, as is common with seven-feet shells, a much better combustion of the fuel is possible; but with a high pressure like 120 pounds it requires a special construction of the furnace, on one of the systems shown by Fig. 129, to withstand the

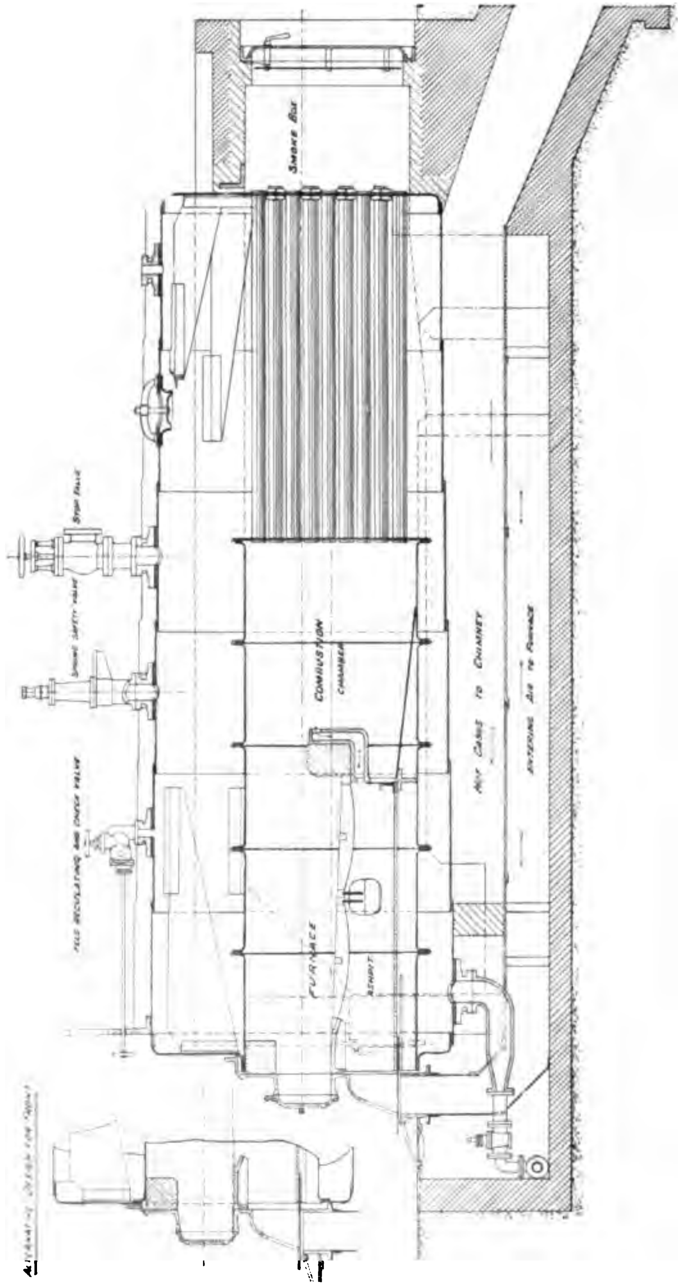


FIG. 134—CORNISH TUBULAR BOILER—LONGITUDINAL SECTION.

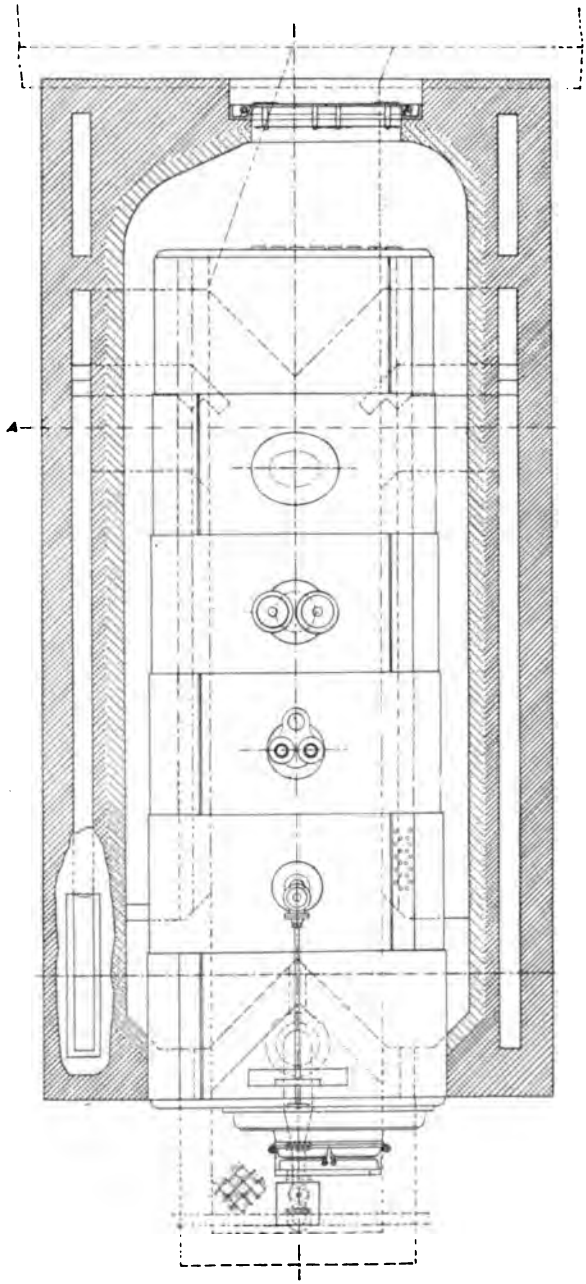


FIG. 135—CORNISH TUBULAR BOILER—PLAN.

collapsing strain with metal of a reasonable thickness. It is the Adamson joint which is shown and adopted, principally for the reason that nearly all first-class boiler shops have now a flanging machine, whereas the various systems of corrugated furnaces require a special plant to produce them. The "Morrison," which has to a large extent superseded Sampson Foxe's original corrugated furnace, is perhaps the most popular one on shipboard now.

THE REGENERATIVE SETTING.

In this boiler, Fig. 134 as well as in the multitubular boiler, Fig. 125, the "setting" is arranged with double flues; those next the boiler itself are traversed by the hot gases of combustion on their way to the chimney while the outer passages in the brick work serve to bring the cold air to the furnace. After several years' experience with boilers set in this way, the author is able to say with confidence that the arrangement is a most successful one. It is really possible to be for some time close at hand to the boiler without knowing it is at work, as the outer brick work keeps perfectly cool. For this reason the walls do not crack and let in the cold air, or require buck staffs to keep them together.

For an ice factory where the water supply is brackish, or has a heavy impregnation of other mineral substances, and space is available, the Lancashire and Cornish tubular boilers may be relied upon for giving satisfaction. Where the water is good the underfired boiler, as Figs. 124, 125, would be the most economical in the long run. Where it is absolutely necessary to make the most steam in the least space, no doubt the locomotive boiler, like Fig. 138, would best answer the requirements. A boiler of this type may be made with tubes as small as one and one-half inches or one and one-quarter inches diameter and by having a forced draft will burn five times as much fuel per square foot of grate surface as would be economical or desirable with either the underfired boiler or the Cornish one.

THE GENERAL CONSTRUCTION AND MOUNTINGS OF BOILERS FOR ICE PLANTS AND COLD STORES.

Leaving for the present the old argument that there is no advantage in having an economical engine to operate the

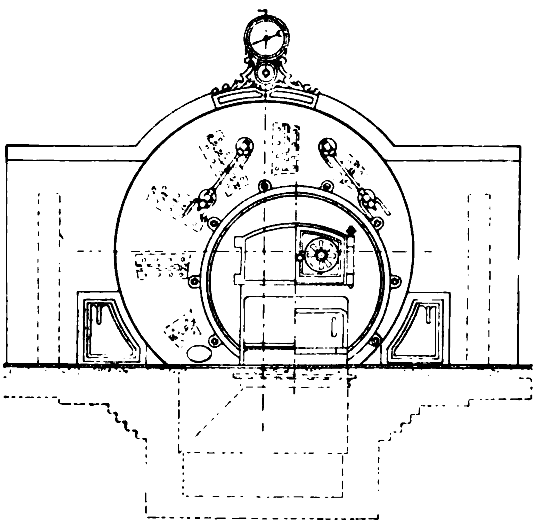


FIG. 136—CORNISH TUBULAR BOILER—FRONT ELEVATION.

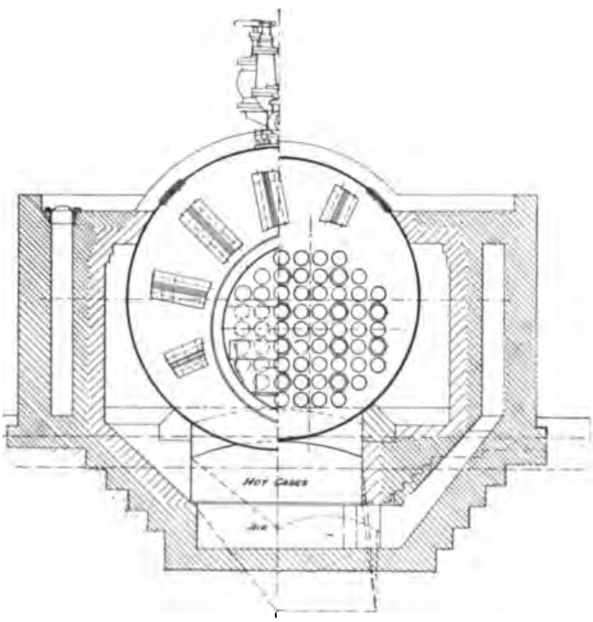


FIG. 137—CORNISH TUBULAR BOILER—SECTION.

machinery in an ice factory, because you must evaporate a greater weight of water to supply the distillate for the ice cans than even a wasteful engine requires, it would be a fair thing to assume a working steam pressure of at least 120 pounds to the inch, if coal costs as much as $2.50, or ten shillings, a ton. If economy based upon the best practice is desired, owing to more costly fuel, then 160 to 200 pounds may be employed. Now in order to carry 120 pounds work-

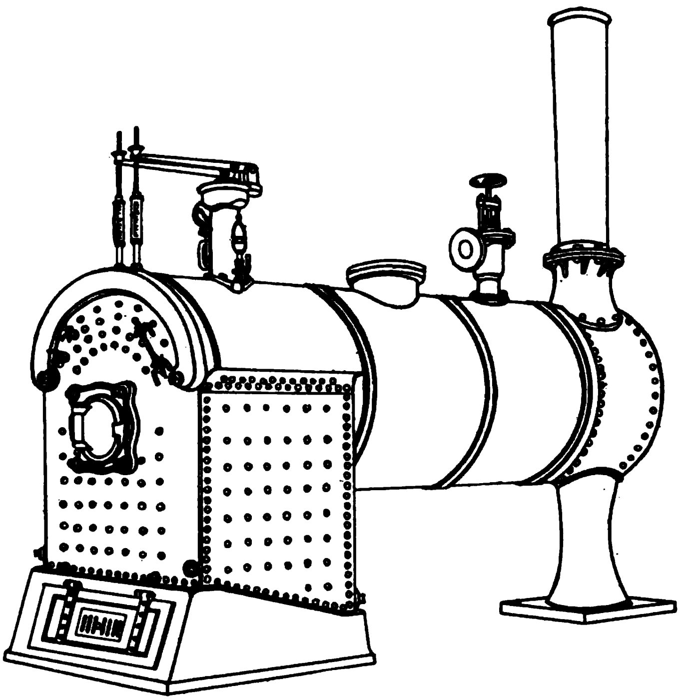
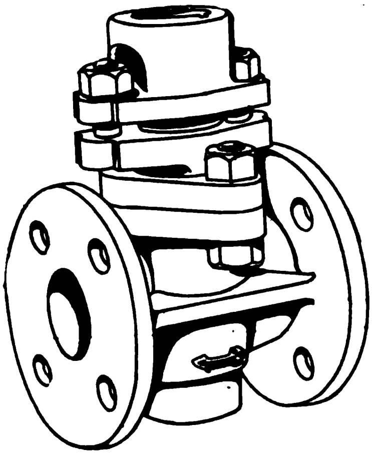


FIG. 138.—LOCOMOTIVE TYPE STATIONARY BOILER.

ing pressure, year in and year out, with satisfaction, the conditions demand first-class material and workmanship, and a cheap boiler will surely prove in the long run a most costly investment; the highest bid however does not necessarily guarantee the highest quality in the article supplied.

Among the many important points to be looked for in a good boiler, the most essential perhaps are among the following: Material, mild ductile steel of moderate—say twenty-eight tons, and not high, say thirty-two tons— tensional strength

under test, for shells, furnaces and flues. All plates to be planed on their edges. All rivet holes to be drilled in place, after the plates are bent. All longitudinal seams to be at least double riveted, or double strapped. Manholes to be strengthened with special reinforcement rings, and have stamped steel corrugated manhole doors. No valves or cocks to be bolted directly on to the boiler shell, but be secured either to solid blocks, as in Figs. 124 and 134, or to short welded steel stand-pipes riveted on, as in Fig. 131. The very best gauge-glass mountings procurable, preferably asbestos packed, should only be used; and where they require pipes as in Fig. 125, these connections should be of copper with screw



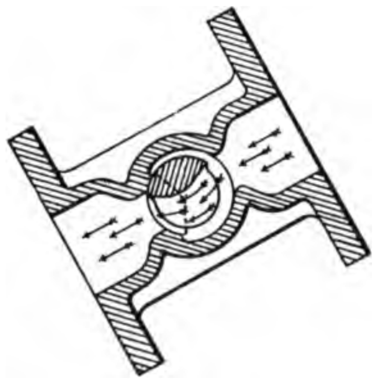


FIG. 139. BJORNSTAD'S BLOW-OFF COCK. FIG. 140.

unions. All underfired boilers to have their bottom set with a fall of two or three inches to a chamber to receive deposit for the blow-off cock or valve. If perforated pipes lying on the bottom are used for blow-off, then a frequent use is required, especially before raising steam, to remove deposit thrown down. Spring or dead weight safety valves to be used in preference to those with levers, which latter often vibrate with the pulsation of the steam flow to the engine. All the stop valves to have external screws. The best blow-off cock yet invented appears to be the one shown by Figs. 139 and 140, which has the following characteristics: There are no "ground" surfaces exposed to the deposit when the

cock is shut, consequently when it is opened there is no scoring caused to make it leak. It can be packed under steam. The key cannot be withdrawn until the cock is completely closed.

Plenty of space should be left around the tubes for their examination and cleaning. Louvres or regulators should be fitted to the doors, to regulate the supply of air both below and above the fire. The fire bars to be specially suited for the grade of coal used and the rate of consumption. All internal furnaces or flues to be strengthened on one of the systems shown in Fig. 129, so as to avoid the necessity for heavy plates, and the risk of burning them.

Above all, let the intending purchaser beware of the "great economy" fiend; and (although it is an old chestnut) it will be well for him not to forget the story of the steam user who adopted all the latest improvements offered to him, and when he had paid all the bills and totted up what had been promised (as is promised every day), he obtained the following as the result of the gross saving to be expected: By contorted tubular boiler, 20 per cent; acrobatic fire bars, 10 per cent; steam dryer, 5 per cent; automatic damper regulator, 5 per cent; patent cut-off, 15 per cent; waterless condenser, 20 per cent; economizer and feed heater, 25 per cent; purifier and softener, 10 per cent, or a total saving of 110 per cent. He therefore thought he should be burning 10 per cent less than nothing, and his coal heap should be getting larger; but somehow or other he found the coal went away just about the same as before.

Lying open on the table as this is being written, is an advertisement, in a highly reputable journal, which boldly undertakes to increase the efficiency of the boiler up to 55 per cent by the adoption of the one particular device offered. Now where things are so bad that a 55 per cent improvement is possible, it may in most cases be taken for granted, that all the saving will not be effected by one piece of apparatus, but will probably require the whole steam plant remodeled by a competent expert. The greater economy which increased steam pressures, and higher grades of expansion will effect, are shown in the following table, which gives the relative quantity of coal required for the same horse power, under

different steam pressures up to 300 pounds to the inch, and with grades of expansion up to eight fold. It must not be forgotten that 180 pounds of steam is now a pressure in common use, both on land and at sea:—

COMPARATIVE WEIGHT OF COAL REQUIRED PER HORSE POWER PER HOUR, WITH STEAM PRESSURES FROM THIRTY TO 300 POUNDS PER SQUARE INCH, AND GRADES OF EXPANSION FROM 0 TO 7/8.

Steam Pressure in Pounds per Square Inch.	Grade of Expansion.								
	0	1/4	1/3	3/8	1/2	5/8	2/3	3/4	7/8
	Weight of Coal in Pounds.								
30	5.6	4.93	3.95	3.81	3.30	2.84	2.69	2.35	1.82
35	5.51	4.84	3.86	3.72	3.21	2.74	2.60	2.26	1.73
40	5.46	4.79	3.81	3.67	3.16	2.70	2.55	2.21	1.68
45	5.41	4.73	3.75	3.62	3.11	2.65	2.50	2.16	1.62
50	5.36	4.68	3.71	3.57	3.06	2.60	2.45	2.11	1.58
55	5.31	4.63	3.66	3.51	3.01	2.55	2.40	2.06	1.53
60	5.26	4.59	3.60	3.47	2.97	2.50	2.35	2.02	1.49
65	5.20	4.55	3.57	3.43	2.93	2.46	2.31	1.98	1.45
70	5.19	4.52	3.54	3.40	2.90	2.43	2.28	1.94	1.41
75	5.16	4.49	3.51	3.37	2.87	3.40	2.25	1.91	1.39
80	5.12	4.45	3.47	3.33	2.83	2.36	2.21	1.88	1.35
85	5.09	4.42	3.44	3.30	2.80	2.33	2.18	1.85	1.32
90	5.07	4.39	3.41	3.28	2.77	2.31	2.16	1.82	1.29
95	5.04	4.37	3.39	3.25	2.74	2.28	2.13	1.79	1.26
100	5.01	4.34	3.36	3.23	2.72	2.26	2.10	1.77	1.23
105	5.00	4.32	3.35	3.21	2.70	2.24	2.09	1.75	1.22
115	4.98	4.31	3.33	3.19	2.69	2.22	2.07	1.73	1.20
125	4.94	4.27	3.29	3.15	2.65	2.19	2.03	1.70	1.17
150	4.81	4.14	3.16	3.02	2.52	2.05	1.90	1.57	1.04
200	4.70	4.03	3.05	2.91	2.41	1.94	1.79	1.46	0.92
250	4.69	3.93	3.01	2.81	2.31	1.85	1.70	1.36	0.83
300	4.54	3.87	2.89	2.75	2.24	1.78	1.62	1.29	0.75

This table shows that with the low pressure of thirty pounds steam, and no expansion, as was common many years ago, the consumption of coal would be double that required with eighty-five pounds pressure and a cut-off at half stroke; and further, that more economy can be obtained by increasing the expansion and raising the pressure, until the consumption is only one-seventh of that given under the lowest conditions.

After having secured a good boiler, the next thing is to have it properly set, with the sectional area of the flues so

proportioned for the volume of the gases to be carried to the chimney, as to get the best results from the fuel. Many arguments are being put forward in favor of a mechanical draft, urged by fans, instead of having the natural draft of a chimney, and some of them are very specious. It would be going outside the general scope of this work to discuss this question in detail, and it may be left with the remark that a tall chimney at any rate carries the heated waste gases away well clear of the factory, which the short stumpy outlets much advocated by some engineers certainly do not, and a chimney certainly wants no attention in comparison with a fan or exhauster.

Having the boiler set with double side walls, and the top above the brick work encased with at least two inches of good non-conducting composition, the whole setting and casing, as well as the house, should be kept scrupulously clean and white; then the radiation of heat will be reduced to a minimum. Black, dirty boilers, and settings smothered in dust, with dark and dirty surroundings, all greatly favor the radiation and conduction of heat, which as before shown, is specially objectionable in an ice factory.

In connection with the efficiency of engines and boilers, no work has probably ever been done of such service to the general steam user—to enable him to see where the losses really occur—as the report and diagram on the Louisville pumping engines recently issued by a committee of the Institute of Civil Engineers.

This celebrated Leavitt engine, at Louisville, has been described in the transactions of the American Society of Mechanical Engineers. Its operations have since been investigated by a committee of the English society appointed in 1896 to establish a standard for comparing and judging the thermal efficiency of steam engines, and has resulted in a report, and the diagram reproduced in Fig. 141.

This figure illustrates the flow of heat, in British thermal units, from the furnace to the actual brake power exerted. The various losses or leakages by radiation, condensation, and so on, are clearly shown; and also the saving of heat again picked up, as by the economizer, and the return of hot water from the jackets.

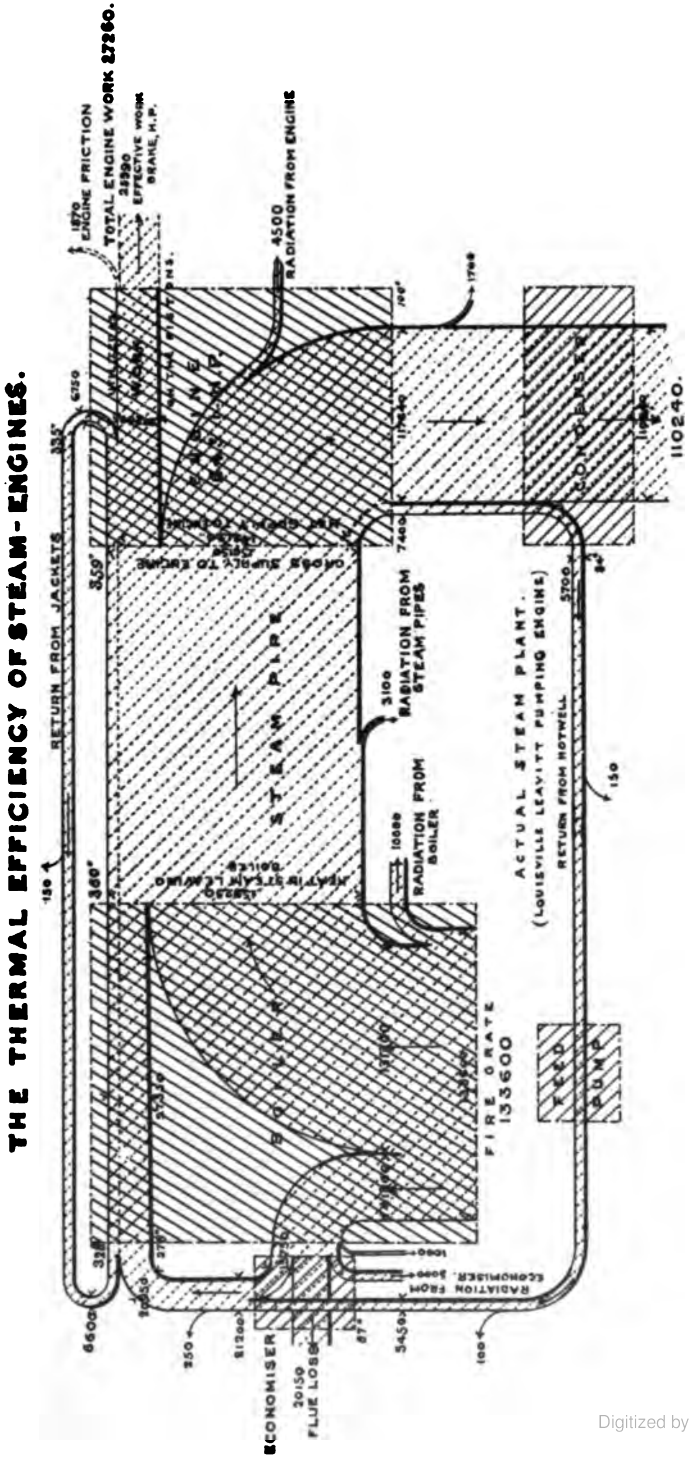


FIG. 141.—DIAGRAM SHOWING THE FLOW OF HEAT IN HIGH CLASS STEAM ENGINE.

The heat put into the water from the furnace per minute is 133,600 units, and that represented by the brake power is only 25,990 units, or say 19 per cent; that lost or thrown away in the condenser alone being 110,240 units, or over 58 per cent. The radiation from the boiler, the steam pipes and the engine is comparatively small, and the flue and other losses are so relatively insignificant, that when an inventor comes along with his offer of 50 per cent saving, the steam user having this diagram in hand, may possibly be able to tell his would-be benefactor that he is professing to save a great deal more heat or power than is actually lost. In the Leavitt engine, 221 units per minute are required for an indicated horse power, which in an ideal engine are reduced to 148 units.

CHAPTER XVIII.

ICE PER TON OF COAL.

Looked at as commercial operations, the success, or otherwise, of both ice manufacture and refrigeration is largely a question of coal consumption. It is no doubt true that instances are common where it is advisable to expend a little extra money on fuel rather than incur the additional first cost and subsequent up-keep which would be involved in the change to more economical machinery and highly refined appliances. At the same time it really does seem—if the records are true—that many ice factories are altogether more wasteful in the use of fuel, and show poorer results, than there is any necessity for.

At the annual meeting of the Southern Ice Exchange of the United States held at St. Louis in 1898, a paper was read in which the author, Mr. Sneddon, gave the results obtained by him from twenty-seven different ice factories. These are reproduced in the table on the following page, and show that the water evaporated, or ice made, per pound of coal, ranged from 8.22 pounds in the best, to only 2.25 pounds in the worst case. This in itself appears a very wide range of relative efficiencies, the better results being more than three and one-half times as much as the poorer ones, and it is made the more singular from the fact that the thermal efficiency of the coal used for the smallest evaporation was fully equal to that used for the highest.

The best of these tabulated cases, however, compares very poorly with the result of some tests which were made at a Bavarian brewery twelve years earlier, and are recorded in a paper read by the managing director of the British Linde Company before the Institute of Mechanical Engineers in

1886. It is there stated, that as much as 26.3 tons of ice have been made for the ton of coal. As this is more than ten times as great as the results in some of the factories referred to by Mr. Sneddon, and as it is impossible that the difference in climate, and temperature of condensing water, can be responsible for the whole of such great discrepancies, it will perhaps be worth while to look a little deeper into this question. There is no serious reason, on the face of it, why some of the factories in the list should not at least treble their efficiency, with fair average plants and modern methods.

TABLE OF ICE PLANT EFFICIENCIES COLLECTED FROM TWENTY-SEVEN EXISTING AND OPERATING PLANTS (SNEDDON)

Ice produced in tons per 24 hours.	Coal consumed in lbs. per 24 hours.	Pounds of water evaporated to 100 lbs. G. pressure from 212° per 24 hours (or max. ice production).	Pounds of water evaporated per lb. of coal (or lbs. of ice made).	B. T. U. contained in one lb. of coal (calculated).	Total heat put into total water evaporated by one lb. of coal.	Efficiency per cent.	Loss on 70 per cent basis per cent.
5.4	4,800	10,800	2.25	13,400	2,261	17.6	75.
5.7	4,800	11,400	2.37	13,400	2,381	17.7	74.8
7.25	4,000	14,500	3.62	12,200	3,638	29.8	57.5
7.5	4,000	15,000	3.75	12,200	3,768	30.8	56.0
10.33	5,000	20,660	4.13	14,858	4,150	27.9	60.2
11.	5,000	22,000	3.93	14,858	3,949	19.9	71.6
14.	12,000	28,000	2.33	12,700	2,341	18.4	74.3
14.5	9,000	29,000	3.22	11,900	3,236	27.2	61.2
16.5	9,500	33,000	3.47	11,900	3,488	29.3	58.0
15.6	9,600	31,200	3.25	12,300	3,266	26.5	62.2
16.5	11,200	33,000	2.94	12,300	2,954	24.	65.8
20.	12,000	40,000	3.33	12,600	3,346	26.5	62.2
19.	8,000	38,000	4.75	12,200	4,773	39.1	44.2
14.5	6,000	29,000	4.83	12,200	4,854	39.8	43.2
17.5	10,000	35,000	3.5	12,000	3,517	29.3	58.0
17.66	10,000	35,320	3.53	12,600	3,547	28.1	60.0
27.5	13,500	55,000	4.07	13,000	4,090	31.3	55.3
19.	7,000	38,000	5.42	12,200	5,447	44.6	36.3
20.	6,000	40,000	6.66	13,000	6.693	51.4	26.6
23.	6,800	46,000	6.76	13,000	6,793	52.2	25.5
24.	7,000	48,000	6.85	13,000	6,884	53.0	25.8
29.	14,000	58,000	4.14	12,000	4,160	34.6	50.6
25.	18,000	50,000	2.77	12,000	2,783	23.2	66.9
32.	22,000	64,000	2.90	12,000	3,045	25.3	62.9
31.	14,000	62,000	4.42	10,500	4,442	42.2	39.8
82.	22,500	164,000	7.28	13,100	7,286	55.6	20.6
85.	20,740	170,000	8.22	13,100	8,261	63.0	10.0

It is evident that the plea, "There is no advantage in having an expansive engine, because you would have to con-

dense live steam to make the distilled water for the cans," does not apply in these cases. It certainly does not require specially good boilers to evaporate six and three-fourths pounds of water per pound of coal, considering that eight to nine pounds is easily attainable. Taking the moderate evaporation of six and three-fourths pounds only, and allowing one-third of it, or two and one-fourth pounds, for waste in condensation, drainage, etc., there would still be four and one-half pounds left to make ice from, or double the amount actually yielded by the plant.

In making ice there are so many minor losses from radiation, conduction, thawing out, and so on, which affect the ultimate result, that it makes it exceedingly difficult to calculate from theoretical data before-hand, what the production of a new plant will come up to. The practical man who has rule-of-thumb notes, deduced from the working of similar plants under varying conditions, will probably get nearer to the mark than the engineer who calculates everything on a scientific basis alone.

The main factors which are concerned in the question of maximum ice for minimum coal versus minimum ice for maximum coal, are as follows:—

First.—There is the thermal efficiency of the coal itself, which may range from 10,000 to 15,000 units in a pound, and the efficiency of the boiler as a machine. Although the best coals are theoretically equivalent to fifteen pounds of water from 212°, the highest evaporation in actual practice does not much exceed ten pounds of water per pound of coal,* while seven pounds is a low result. It will be a fair thing to assume eight and one-half pounds as a fair average evaporation attainable in an ordinary factory.

Secondly.—There is the efficiency of the steam engine in terms of the weight of steam consumed. The very highest result so far published, appears to have been attained by an experimental quadruple-expansion engine at the Cornell University, with a boiler pressure of 500 pounds to the square inch, and a record of ten pounds of steam per horse-

*See reference to boiler, Fig. 135, page 219, result of trial.

power-hour. Nothing like this is possible in actual work at present, and the very best marine engines probably do not use less than thirteen pounds, even when working with triple expansion and an initial pressure of 200 pounds of steam. The following is perhaps a fair average of steam consumption in commercial, as distinguished from experimental, engines:—

	Pounds of steam per horse power per hour.
Condensing, quadruple and triple expansion..	14 to 16 lbs.
Condensing, compound........................	16 to 24 lbs.
Non-condensing, compound or expansion......	20 to 30 lbs.
Condensing, low pressure....................	30 to 40 lbs.
Non-condensing, low pressure................	40 to 60 lbs.

If a plant of machinery is intended for refrigeration only, and distilled water is not required, there are no absolute reasons why the steam engine supplying the power should not have a surface condenser, and if on shipboard, be worked at the same pressure, and with the same degree of economy, as the main engines. In such case an indicated horse power might be obtained by the expenditure of from 1.5 to 1.7 pounds of coal.

When however distilled water must be had in order to make clear, crystal, can ice, it will be better to work the engines non-condensing, and with a back pressure, under one of the two systems to be presently described, and to use a high initial steam pressure with a high grade of expansion, preferably in a compound or triple expansion engine. Although the increased range of temperatures due to a vacuum will be sacrificed so far as the engine is concerned, by not expanding down below atmospheric pressure, there will be no difficulty even then in getting an indicated horse power with twenty-five pounds of steam per hour. The condenser and vacuum, as will be seen later on, can be turned to better account than simply to increase the power of the engine.

Thirdly.—There is the efficiency of the compressor as a complete machine, or the ratio which the indicated horse powers of the steam cylinder, and the compressor, bear to one another. The following table has been collated from the several examples therein quoted, and it shows that the frac-

tional losses in such machines range between 12 per cent and 33 per cent of the total engine power:—

Authority or source of the information.	Horse power of the engine.	Horse power of the compressor.	Ratio of compressor to engine power, per centum.	Friction or loss in terms of the engine power, per centum.	Friction or loss in terms of the compressor, per centum.
Mr. A. Siebert in *Ice and Refrigeration* for January, 1899	......	......	......	25 % to 33 %	33 % to 50 %
Diagrams illustrating their machines from the De La Vergne catalogue	63.0	48.0	76.1%	23.9%	31.4%
Bavarian brewery in 1886.	53	38	71.7	28.3	39.4
Comparative trials in 1890, Linde and Pictet machines—					
Average of four Pictet.	......	......	79.3	20.7	26.1
Best Pictet trial	......	......	81.1	12.9	14.8
Average of four Linde.	......	......	83.4	16.6	19.9
Best Linde trial	......	......	87.9	12.1	13.7
"Eclipse" machines— Frick Co.'s Red Book illustrations	60.3	51.6	83	17	20.4
"Case" machine from the company's book	63.9	56.5	88	12	13.6
Case Co.'s guarantee	......	......	87	13.04	15
Fair average to assume with a good design and the best workmanship.	100	83.3	83.3	16.6	20

It will be seen from the foregoing table that the highest efficiency is obtained with a machine which has its steam and ammonia cylinders connected up in a straight line, fully supporting what was said in previous chapters, as to frictional losses by round-about connections. The makers of

this machine guarantee that their engine power will not exceed the compressor power by more than 15 per cent. It will leave a considerable margin, if in considering the whole question of efficiency, we assume the indicated engine horse power in a new plant at 20 per cent in excess of that of the compressor.

Fourthly.—There is the efficiency of the compressor itself considered as a pump, which may vary between very wide limits. In the year 1878, the writer designed the compressors, reservoirs, and reducing valves, that have been successfully used ever since that date, for lighting the cars on the New South Wales railways with gas. The original machinery was all made in Sydney, and the pressure was intended to range between 120 and 180 pounds. A large imported compressor was subsequently put to work, which, when tested by him, was found to deliver only about one-half of its theoretical capacity, the defects being due probably to small valves, too large clearance, and the great heat generated.

Although some makers claim 98 per cent efficiency for their own manufactures, such machines will be very effective, and have small clearance, if they can be kept so cool as to pump 95 per cent of their theoretical volume from the refrigerator. Unless frozen well back, 90 per cent would probably be nearer to the average effect obtained.

Then there is—*Fifthly.*—The height which the abstracted heat has to be lifted from the temperature in the refrigerator, in order that it may be carried away by the condensing water.

From Munich to Central Australia is a far cry, and these places present very different conditions for the ice maker to study. In the records of the Munich experiments the temperatures of the condensing water are given as :—

At entrance	49°	49°	48°	48°	Fah.
At exit	67°	67°	49°	67°	"

In a machine designed by the writer for an East Indian city it was stipulated among other conditions of the trial, that the average atmospheric temperature was to be 95° and the water 90°. In some towns in Australia, such as Bourke, out west (where the crust of the earth is said to be very thin between the people and the place below where there is no

ice), the summer heat is often 120° in the shade for long periods together.

As far as the time required for freezing the blocks of ice is concerned, the brine might be kept at the same temperature at these two places having such extremes of climate, but with 40° difference in the surroundings, the greater leakage of heat through the insulated walls of the tank would cause a much more serious loss in the hot climate. Further, the greater tendency of the ice to thaw when drawn from the cans might be an inducement to freeze colder in the hot than in the cool city. Such conditions would widen the disparity in the relative efficiencies of the two plants, when measured by the weight of ice produced for sale at such widely separated localities, from a given weight of fuel.

In order to make a comparison of the relative work required to make ice in the two cases, we may omit the latter considerations and take a back pressure of twenty-four pounds (gauge) or thirty-nine pounds (absolute) in both climates, with a condenser temperature of 65° in Bavaria and of 105° in Central Australia, the gauge pressures being 103 pounds and 218 pounds respectively. The table on following page gives the volume of gas required to be pumped per minute in cubic feet to produce one ton of refrigeration, and for purposes of comparison these quantities will be doubled as is usual to give the amount per ton of ice. This leaves a considerable margin for waste, and much more in the case of the cold climate, because forty more thermal units have to be abstracted to make ice from water at 90° than from water at 50°, while the melting of a ton of ice represents an absolute quantity of heat or work in any climate.

The table under column headed 24 (as the gauge suction pressure) shows, that with a terminal pressure of 103 pounds to the inch, the gas required to be withdrawn from the refrigerator will be 2.87 cubic feet per minute per ton of refrigeration; and with 218 pounds terminal pressure, then 3.12 cubic feet per minute must be withdrawn.

Now by plotting the isothermal and adiabatic lines of compression, from thirty-nine pounds to 118 pounds absolute pressures, to represent the work in a cool climate, and

CUBIC FEET OF GAS PER TON OF REFRIGERATION, WITH REFRIGERATOR AND CONDENSER UNDER DIFFERENT TEMPERATURES AND PRESSURES.—[From *Compend of Mechanical Refrigeration.*]

Suction pressure by gauge in lbs. per square inch......		1	4	6	9	13	16	20	24	28	33	39	45	51
Corresponding temperature in refrigerator in deg. Fahr..		—27°	—20°	—15°	—10°	—5°	0°	5°	10°	15°	20°	25°	30°	35°
Condenser pressure (by gauge) lbs. per square inch.	Corresponding temperatures in degrees Fahr.	Number of cubic feet of ammoniacal gas required to be compressed per minute to produce one ton of refrigeration in 24 hours.												
103	65°	7.22	5.84	5.35	4.66	4.09	3.59	3.20	2.87	2.59	2.31	2.06	1.85	1.70
115	70°	7.3	5.9	5.4	4.73	4.12	3.63	3.24	2.9	2.61	2.34	2.08	1.87	1.72
127	75°	7.37	5.96	5.46	4.76	4.17	3.66	3.27	2.93	2.65	2.36	2.10	1.89	1.74
139	80°	7.46	6.03	5.52	4.81	4.21	3.70	3.30	2 96	2.68	2.38	2.12	1.91	1.76
153	85°	7.54	6.09	5.58	4.86	4.25	3.74	3.34	2.99	2.71	2.41	2.15	1.93	1.77
168	90°	7.62	6.16	5.64	4.91	4.30	3.78	3.38	3.02	2.73	2.44	2.17	1.95	1.79
184	95°	7.70	6.23	5.70	4.97	4.35	3.83	3.41	3.06	2.76	2.46	2.20	1.97	1.81
200	100°	7.79	6.30	5.77	5.05	4.40	3.87	3.45	3.09	2.80	2.49	2.22	2.00	1.83
118	105°	7.88	6.43	5.83	5.08	4.44	3.91	3.49	3.12	2.82	2.51	2.24	2.01	1.85

from thirty-nine to 233 pounds, for the requirements of nearly tropical surroundings, the following Fig. No. 142 is the result.

We find from the above that the mean pressures during compression are forty-seven pounds and 78.5 pounds under the two different conditions. Now 78.5 pounds×144 inches= 11,304 foot-pounds as the amount of work required to compress one cubic foot of gas, and 11,304×3.12 cubic feet gives 35,268 foot-pounds per minute as the work for one ton of re-

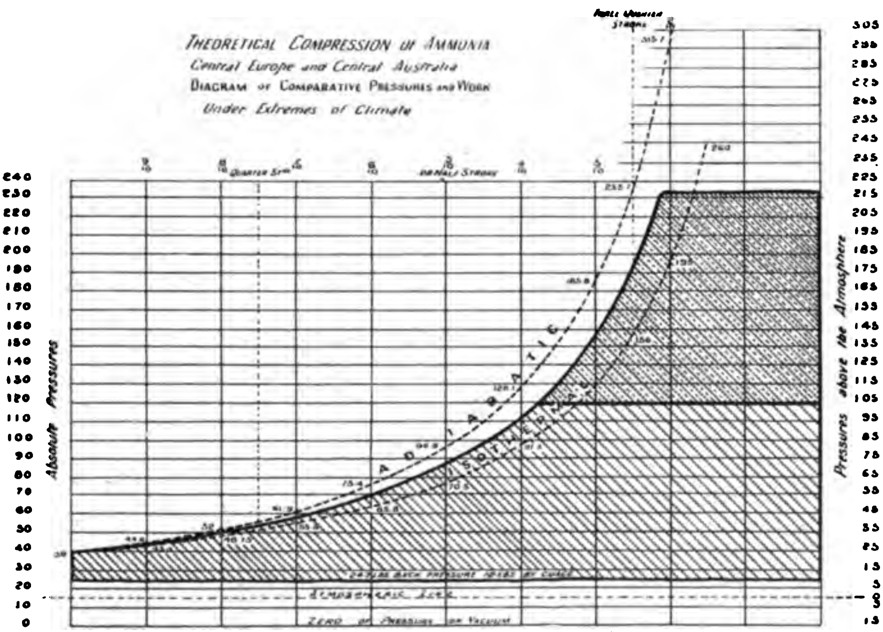


FIG. 142.—THE WORK OF AN AMMONIA COMPRESSOR AS AFFECTED BY CLIMATIC CONDITIONS.

frigeration in twenty-four hours; which divided by 33,000 gives the horse power of the compressor. Similarly, 47 pounds×144 inches=6,768 foot-pounds, which multiplied by the 2.87 cubic feet required, gives 19,424 total foot-pounds of work, necessary at the lower temperature of condenser, per minute per ton of refrigeration. Then $\frac{19{,}424 \times 35{,}268}{2} = 27{,}346$ as the power required for a mean between the two extremes of climate.

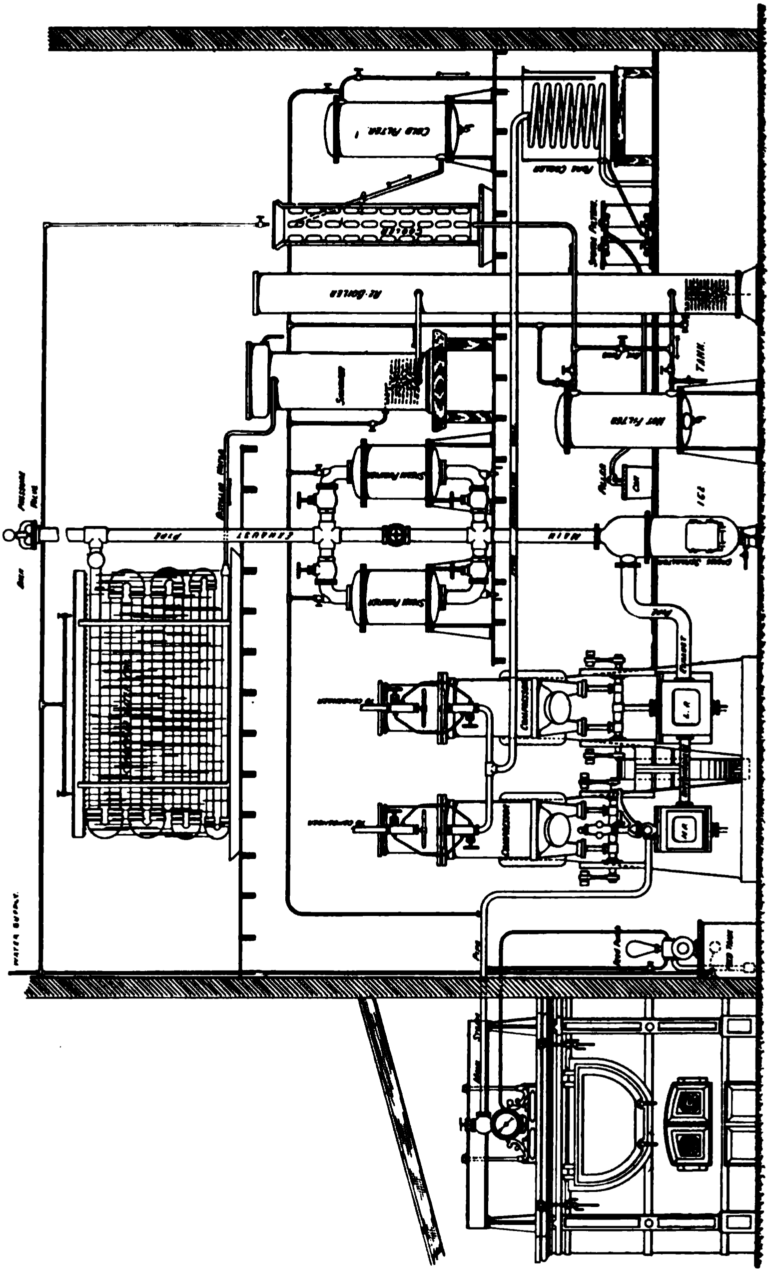


FIG. 143—ICE FACTORY CONDENSING AND PURIFYING EXHAUST STEAM FOR FILLING THE ICE MOLDS.

Let 27,346 foot-pounds per minute in the compressor under a mean temperature be the equivalent to one ton of refrigeration, and doubling the same, take 54,692 foot-pounds as equal to the work of making one ton of ice; then adding 20 per cent for frictional losses, we get 65,630 foot-pounds per minute as the work of the engine, which divided by 33,000 gives two horse power, very nearly, to make one ton of ice under the mean of the two climates.

Going back to the hottest condenser again and taking 35,268 foot-pounds of work in the compressor per ton of refrigeration (instead of 27,346, the mean of the two climates), then $35{,}268 \times 2 + 20$ per cent $= 84{,}643$ foot-pounds, and $\frac{84{,}643}{33{,}000} = 2.56$ horse power per ton of ice required for the hot climate. In the trade lists of a number of leading manufacturers of refrigerating machinery it will be found that about 2.3 horse power per ton of ice is set down as the power of the engine, and as it is deduced from experience, it is probably a very fair average.

Now 2.3 horse power, consuming twenty-five pounds of steam per horse power per hour, only amounts to $25 \times 2.3 \times 24 = 1{,}380$ pounds of water a day; but a ton of ice (American)=2,000 pounds, about half as much again, and this is apart from waste. Therefore if the ice is to be made from distilled water, that taken from the exhaust of the engine would be insufficient, and it would be necessary to provide boilers (as usually recommended by builders) at least 50 per cent greater horse power than the engines, and condense the live steam. Adding one-half more to 1,380 pounds (or 690) gives 2,370 pounds evaporation required from the boiler for each ton of ice, and this leaves a fair margin for loss and waste in filling molds. Where however there is an engine very wasteful of steam, the exhaust alone may be depended upon for supplying the distilled water to make clear ice.

Fig. 143 shows one of the utterly unscientific and dirty ways in which this process is usually carried out; and as it represents in the main an outfit designed by the writer himself, it is to be hoped that makers who supply similar plants

will not feel hurt at its being spoken of in such terms of disparagement.

This figure being intended to illustrate the several processes carried on generally, rather than the relative proportions of the various vessels (concerning which great diversity of opinion exists), it is not drawn to scale as a working drawing.

As a typical representation of a very common class of plant for producing distilled water for ice making, the operation of its various parts to that end may be traced. First, the boiler. This is generally supplied with unfiltered water, and primes more or less, carrying dirty particles over into the engine. Sometimes there are constituents in the water which give off gases as its temperature is raised, and such gases pass into the engine with the steam.

In the engine, the steam meets with the oil which is used to lubricate the cylinder; and often, owing to poor workmanship, bad alignment, rough boring, and strong springs in the piston rings, a great deal more oil is used than would be otherwise necessary, and the cylinder and piston are gradually ground away. As a result the impure steam, the oil, the abraded iron from the compressor, and other foreign matters from the boiler, all form a delightful composition (!) which passes off by the exhaust pipe, to be converted into pure (?) distilled water for the ice cans.

The first process to which this exhaust is subjected, is intended to get rid of the heavier matters and the grease, and an apparatus for this purpose is shown on the figure as the grease separator. So many of these appliances are now on the market—being illustrated in the current engineering journals—that it is not necessary to describe any particular one in detail, but as the separator in the figure is shown with a door, which is not common, it may be explained that such arrangement facilitates the employment of coke or pumice, if any such material is used to absorb and take up greasy matters in addition to the ordinary deflectors, whether spiral or otherwise, which are used to precipitate them.

Flowing upward, the exhaust steam (which is kept at a slight back pressure by a loaded valve on the summit of the

main pipe) is further cleansed by the two vessels called steam purifiers.

The main central pipe has a by-pass valve between the inlet and outlets, by shutting which either one or both purifiers can be used, and by means of the inlet and outlet valves, one purifier at a time can be cut out for cleaning.

Whether cut straw, coke, pumice, broken bricks, or any other special material is used to take up the impurities depends largely upon the experience of the makers of the plant, or the man in charge after the makers have handed it over. After this process the steam is supposed to be clean enough to be condensed.

The condenser seen on the figure, which has the diameters of the pipes tapering down as in the worm of a spirit still, must not be taken as illustrating ordinary ice making practice. The form of this condenser, and whether it is atmospheric or submerged, depends upon the special circumstances of the case, and although the tapering pipes are theoretically correct, equal efficiency can be attained with ordinary tubes at less cost.

In any large installation of machinery using a steam engine, an exhaust steam feed water heater should certainly be interposed on the feed pipe between the feed pump and the boiler, because it effects a double economy. In the first place by imparting heat to the feed water, coal is saved; and secondly, by abstracting heat from the steam, and condensing some of it in the heater, water is saved, as less condensing water is required to abstract heat at the condenser. Figs. 144 and 145 show a feed water heater (a modification of the well known "Berryman" type) as designed by the author for use with very bad mineral water.

There are several features in this design which distinguish it from ordinary feed water heaters; first there is the connection of every copper tube to the tube plate by gun metal "unions" in such a way that each "U" tube can be separately removed for getting at the deposit and scaling; and further, all the steam and feed branches are kept absolutely clear of the "dome." By this arrangement the outer casing or dome can be lifted at any time by using the by-pass valves and the interior be thoroughly examined and

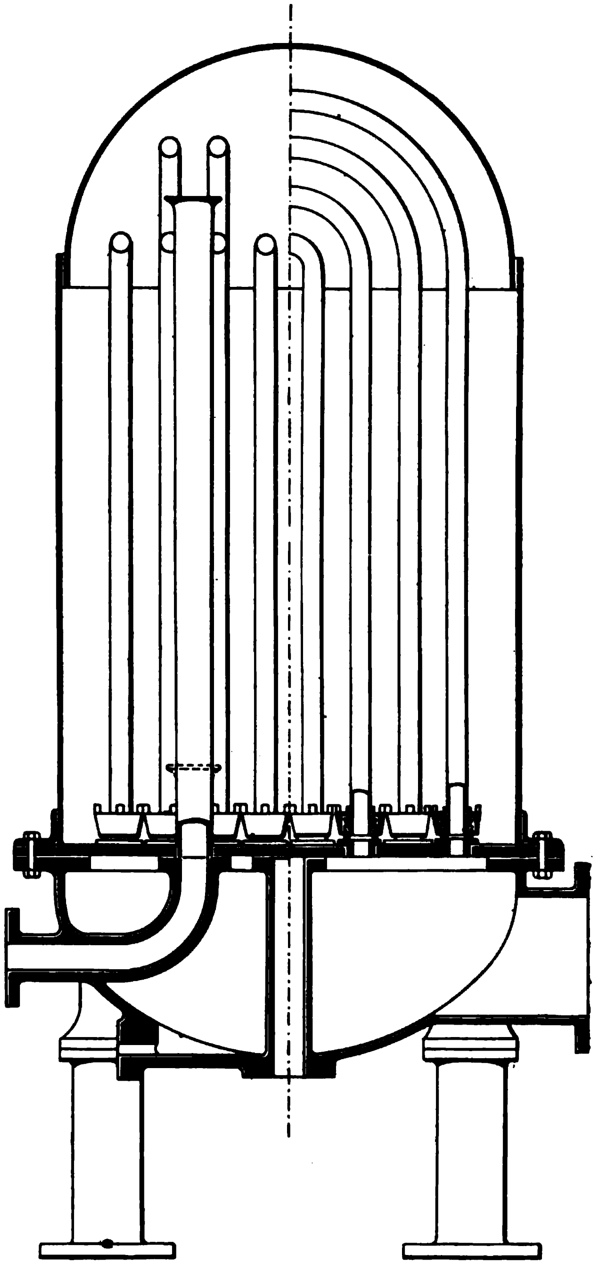


FIG. 144.—SECTION OF FEED WATER HEATER ON LINE *A B* OF FIG. 145.

cleaned, without it being necessary to disconnect a single joint of either the exhaust or feed pipes. The outlet branch to the boiler has an internal stand pipe, so that the heated water is taken from near the top of the vessel and a thorough circulation insured.

In some cases the earthy deposit from bad water can be thrown down at the boiler temperature, without evaporation; in such cases a "live" steam feed water heater such as Figs. 146 and 147 will enable the same to be intercepted before the water reaches the boiler. At the same time the heating of

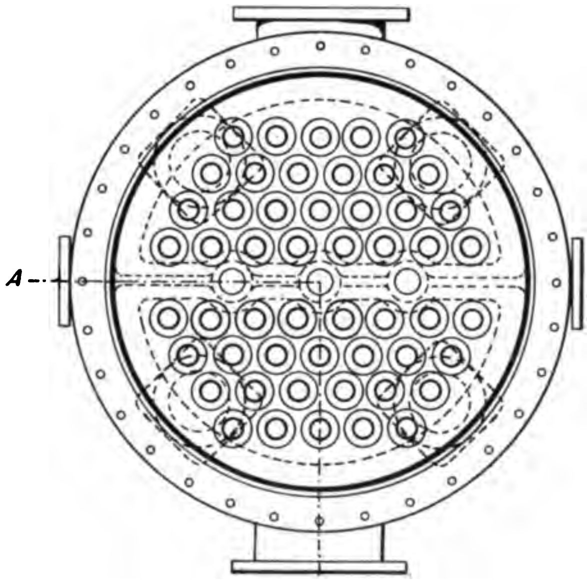


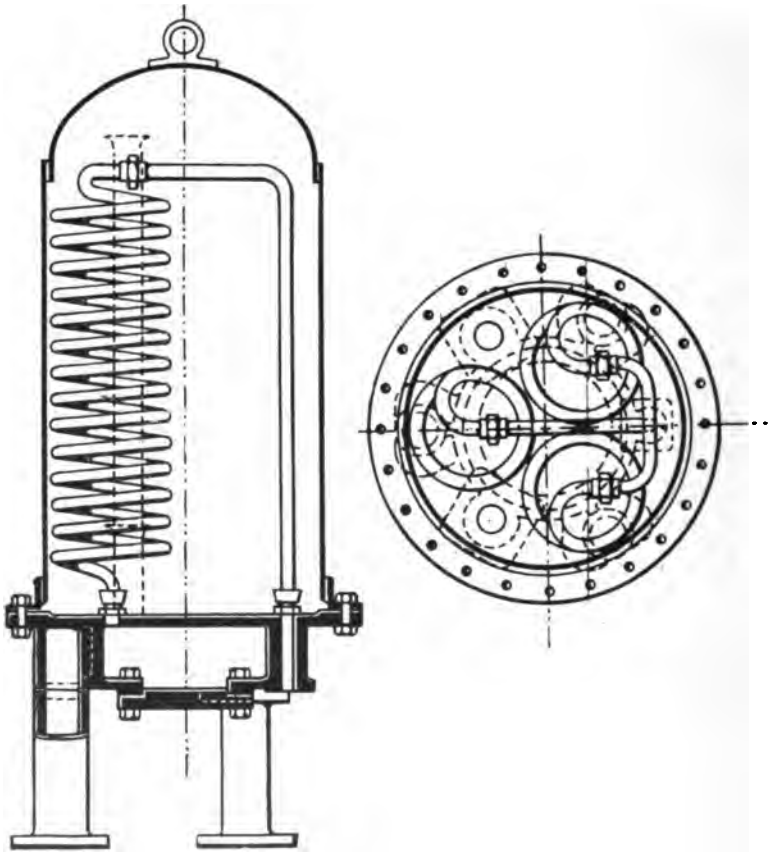
FIG. 145.—PLAN OF FEED WATER HEATER.

the feed will improve the circulation and through that the evaporative efficiency of the boiler. The condensation of steam in the coils of this apparatus produces distilled water, which can either be returned to the boiler by gravitation, or be used for the ice molds. These feed heaters are not shown on Fig. 143, because they are not essential to the purification of the exhaust steam, and the producing of distilled water.

While the exhaust steam feed heater is essential to every plant with a pretense to economy, the supplemental live steam heater has a more efficient substitute, so far as

economy is concerned, in an "economizer" placed in the flues in order to heat the feed water circulating through its tubes by the waste gases of combustion on their way from the boiler to the chimney.

Coming back to the ice factory which we left at the worm or condenser, the distilled water from the exhaust at



FIGS. 146 AND 147.—LIVE STEAM FEED WATER HEATER.

the tail pipe is never absolutely pure, and may carry with it oily particles which have been vaporized and again condensed. It is therefore a very common practice to again boil up this distillate in a vessel called a "skimmer," which is provided with a circular lip at the water level, over which the lighter matters flow into an annular channel, as they rise to the

surface. The water can be kept in ebullition by a small steam coil in the bottom of the vessel.

As this process does not always get rid of all the air, which is the main cause of the cloudy appearance of can ice, or of other gases that may have an affinity for the water, it is next passed to the "re-boiler," which is shown as a very tall vessel. This piece of apparatus, like the skimmer, has a steam coil in the bottom to maintain the water in ebullition; and it is a matter of strong faith with some engineers, although the opinion does not seem to be a general one, that the height or depth of this vessel, and the consequent additional pressure on the water in the bottom of the same, is a great factor towards securing the expulsion of the air.

From near the bottom of the re-boiler the "solid" de-aerated water passes, hot as it is, to the bottom of the hot filter, and by an upward flow is further purified.

There is room for so much difference of opinion as to the methods and materials of these filters that no opinion can be given as to the best for all cases. What may be best in one country might not be obtainable easily in another place. In many places silica is used for the hot filter.

From the upper part of the hot filter the distilled and filtered water ascends through the coils of a condenser or cooler, often parting with its heat to water which has already done duty on the ammonia condenser. It is then passed through the cold filter and after such second filtration it is ready for the fore-cooler. The cold filter is generally a charcoal filter, and where the maple grows the charcoal made therefrom has a great reputation. The whole theory of filters seems to be now in a transition state, and many authorities hold that a large proportion of them actually contaminate the water that passes through them by favoring the development of the microbe; one thing seems to be certain, apart from first cost the "Pasteur" porcelain filter is the best and most effective for securing a high degree of purification.

As this work is devoted rather to the machinery of refrigeration, than to a discussion on debatable points in the methods usually followed in operating it, no general opinion can be given as to the saving of power or otherwise, that is effected by the introduction of a fore-cooler to bring the dis-

tilled water down towards the freezing point before filling the ice cans with it. From a purely thermo-dynamic aspect it may be argued perhaps, that there can be no saving of power in such an operation, and that more surface in the refrigerator coils would enable the same amount of heat to be transferred from the water to the ammonia through the intervention of the brine. And further, it may be held that even admitting you can raise the ammonia to a higher temperature in the coils of the fore-cooler than you could do in the refrigerating coils, you at the same time increase the volume of the gas the compressor has to handle, and so neutralize all the supposed gain.

Leaving the question whether power is saved by use of the fore-cooler or not, it is certain that the distilled purified and de-aerated water from the cold filter is prepared to take up air from the atmosphere above its surface, just as dry air would take up water when the two are exposed each to the other's influence; and, as air and water have not the same affinity for one another at low temperatures, there may be incidental advantages in cooling the water as low as possible, short of freezing, directly it leaves the cold filter.

Although no "lagging" is shown on the fore-cooler, it should in practice be carefully insulated to prevent the infiltration of heat. From this vessel the chilled water passes through a filter of sponges to remove the last traces of foreign matter, for dirt and dust will get into the fore-cooler tank in spite of a good cover. The writer was once told by an Australian ice man, who had a high reputation for his pure crystal blocks, that he attributed his uniform and successful quality largely to the fact that he personally looked to the cleaning of his sponges every morning. These filters being in duplicate, one can be cleaned at a time without disturbing the continuity of the operations in the filling of the cans.

In some ice plants the return gas pipe from the refrigerator coils to the suction branch of the compressor (which in the illustration, Fig. 143, is shown in series with the coil of the fore-cooler tank) is also passed through another exchanger, for the purpose of cooling the liquid ammonia

from the condenser on its way to the expansion valve and the refrigerator coils.

By this device colder liquid goes to the refrigerator to be evaporated, and is thus able to take up more heat, and warmer gas goes to the compressor. When the writer first saw this plan in operation in Australia about the year 1882 he wrote and asked for an opinion about it, to the chief engineer of a large New York refrigerating machine company, which professed at the time to have the largest experience in the business in the world, and to be the makers of the greatest number of machines. The answer was, "You might as well try to lift yourself by your boot-straps as to try and do any good that way." This not very encouraging reply, or rebuke, did not hurt anybody's feelings, and was not taken as a "settler" even if it was so meant. The fact is, tugging at boot-straps may set your boots more comfortably on your feet, and do good that way, even if it does not enable you to lift yourself off the ground. It is not always the cock-sure man that knows the most.

Proposals which come under the "Robbing Peter to pay Paul" category may be derided by people who, having some practical experience, and a casual acquaintance with theory, think they know everything; but as it is only forty years since the writer was first introduced to refrigerating machinery, he is fully aware that there is a great deal yet for him to learn. Therefore he will refrain from expressing any definite opinion as to the absolute advantages which attend the use of either the fore-cooler for the ice water or the temperature exchanger for the liquid ammonia. It should not be forgotten that there are often incidental advantages in doing what may *prima facie* be looked upon as useless. Even by taking money out of one pocket and putting into another, a man may so alter his balance as to be enabled to walk more upright in the sight of his fellow men.

So far the ordinary process of treating water for can ice making has been described as far as the sponge filters; from these a hose, terminating in a small apparatus called a can filler, enables the ice molds to be filled to a uniform depth with the minimum of attention. It is important that this water should run into the cans without any agitation which

would assist it to re-absorb air. The filler therefore delivers it at the bottom of the can, the water rising slowly and steadily around until the supply is cut off automatically at the right point.

The can filler is a branch pipe on the end of the hose, made preferably of tinned copper, and of sufficient length to reach to the bottom of the ice mold. In the type shown by Fig. 148 the attendant opens a valve at the bottom by pressing a small thumb lever that is retained by a catch; when the water rises to the adjusted height, a float sliding on the main pipe rising also, releases the catch, and the valve closes. In

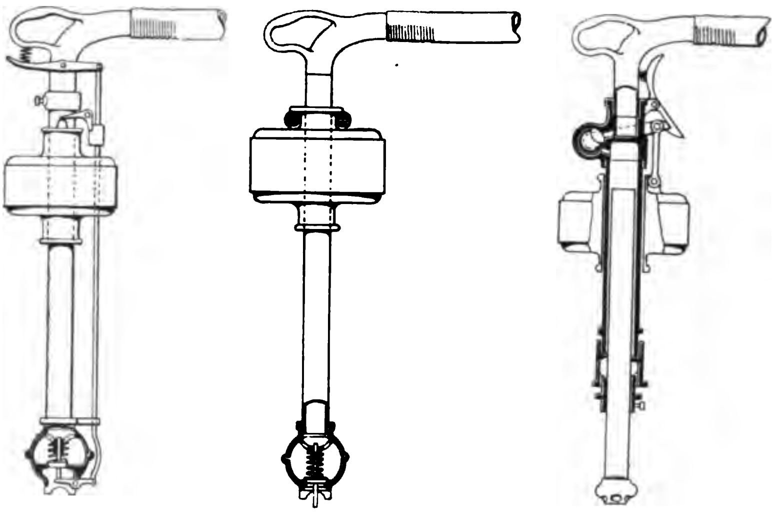


FIG. 148. FIG. 149. FIG. 150.

AUTOMATIC CAN FILLERS FOR ICE MOLDS.

the type shown by Fig. 149, the weight of the pipe resting on the bottom of the can, forces up the valve and allows the can to fill. The float being fixed to the pipe, lifts it up bodily when the water rises, and allows the valve to close and shut off the supply. Fig. 150 shows an Australian filler with a telescope pipe to suit different depths of cans.

The bottom has a bird fountain arrangement, to retain the water in the pipe when it is lifted out of the can, and the upper part has a free working ball-cock, retained by a catch, that shuts off the water when the can is full.

Under the system of distillation so far described, the quantity of distilled water produced, is strictly limited by the weight of steam used by the engine; which, as shown on page 241 need not be more than 1,380 pounds for every 2,000 pounds of ice made. It is therefore usual to evaporate an additional 50 per cent of water by an expenditure of so much extra coal, and then condense the live steam produced, without obtaining any work from it. It will also be noted that the dirty and greasy steam obtained from the exhaust of the engine has to undergo at least eight separate treatments before it is fit to fill the cans to make clear ice from.

Before leaving Fig. 143 it will be noted that many of the vessels have steam connections to their upper ends and purging cocks at the bottom; this arrangement allows them to be cleansed by being blown through. In the case of the filters, arrangements of pipes and valves for reversing the flow of water through them in order to wash them out, should always be supplied and fitted up.

Questions that naturally arise out of the consideration of such a system are: Can the extra expenditure of coal involved in the additional evaporation be dispensed with? Can the process described be superseded by a better one? The answer to both of these is "YES."

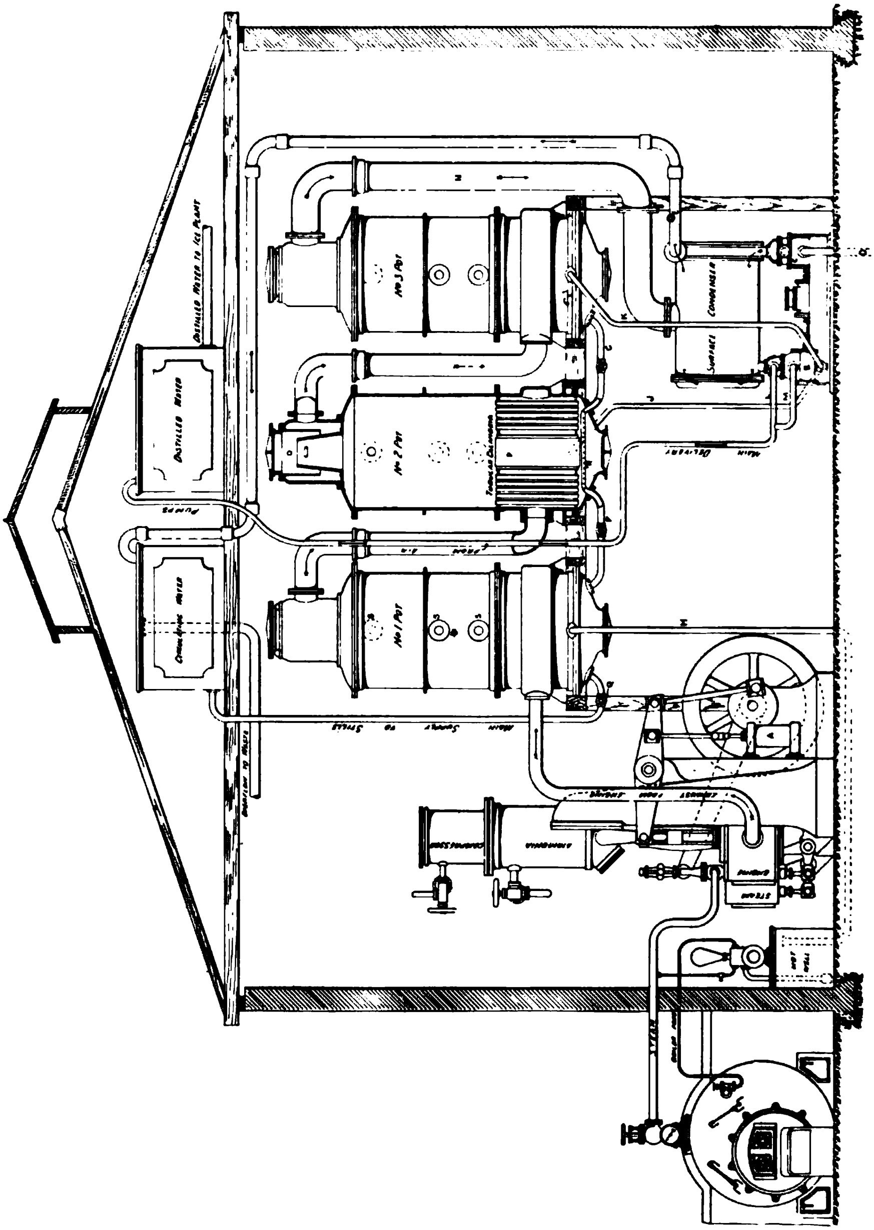


FIG. 151.—ICE MAKING PLANT WITH TRIPLE EFFECT FOR DISTILLED WATER.

CHAPTER XIX.

PURE DISTILLED WATER FOR ICE MAKING.

The evaporation of water for other purposes than raising steam for power is essential to the operations of many industries besides that of making pure ice; but in no other branch of mechanical engineering perhaps has it received so much attention, or been brought to such perfection and economy, as in connection with the work of the sugar refinery.

The system of "double effet" evaporation seems to have had a French origin as the general retention of the French pronunciation would indicate, but there seems to be no good reason why Anglo-Saxons and their kin should continue to say "doobl-affay" instead of "double effect." This system, since expanded to treble and multiple effect, is now so systematized, that with three effects the initial evaporative work of the boiler can be increased two and a half times, and by a multiple system, at least four times the initial evaporation can be secured, without additional fuel.

For ordinary ice factories it is probable that a double effect plant would be ample to produce all the distilled water which a high class economical steam engine could freeze, and with a low class engine a single effect would be powerful enough. It will afford a better illustration of the process however, and produce a larger supply of distilled water per pound of steam with the triple plant to be described.

Fig. 151 shows an elevation of an ice making plant fitted with triple effect evaporators, capable of supplying 2.4 pounds of independently distilled water for every pound weight of condensed exhaust steam from the engine, the water from which can be returned to the boiler.

On the left of the illustration is seen a Cornish boiler, with a lop-sided furnace to facilitate circulation, inspection,

and cleaning. The steam passes by main steam pipe to the compound cylinders of a pair of compressors, which are arranged vertically on the "straight line" system, that is, having direct connection to the engine pistons. The crank shaft and fly-wheel are placed at the back of the compressors, and are operated by beams or levers, through links to the cross-heads and connecting rods. This arrangement reduces the height of the machine, and gives great facilities for the working of either air, circulating, feed or any other pumps—even to deep well pumps—from the levers, should such be required with a plant.

So far there would be no departure from the system shown by Fig. 143. (It may here be noted that the air pump marked A is only an alternative one, and that in such position it would probably take less power to drive it than the direct acting air pump placed under the condenser requires.) Instead of the exhaust pipe from the engines being led through filters and purifiers to an ordinary condenser (and so involve the necessity for a special supply of condensing water, that would afterwards run to waste), it is led into the first vessel marked with the rather unpoetical, although French, title of "pot," where it passes by an annular chamber into the space surrounding or enclosing a number of copper tubes, which tubes are secured in an upper and a lower tube plate. This tubular heat exchanger is called a "calandria," and the illustration shows three pots, each fitted with such a calandria or steam space. The large tube in the center of the space, about eight inches in diameter, is intended to promote circulation in the water to be evaporated.

The water spaces of the three pots are connected at their bottom ends by internal perforated pipes, and also to a source of supply for the water to be evaporated. By adjusting the cocks Q, F and G, the proper water level is maintained in each section while the apparatus is at work.

The last, or No. 3 pot, is in connection with a surface condenser fitted with an air pump, and a high vacuum is maintained in it by the pipe N. The calandrias of the pots 2 and 3 are connected to smaller supplementary pumps by the pipes J and K to draw off the water condensed. A moderate vacuum is maintained in No. 2 and a low vacuum in

No. 1. In the case of No. 1, the exhaust steam is at a back pressure above the atmosphere, and thus the condensed water from the exhaust steam of the engine will flow to the hot well from that calandria without the help of a pump, by the pipe H, for re-delivery to the boiler.

The connections so far being grasped, it will be easily understood that owing to the latent heat, and some of the sensible heat, of the exhaust steam being transferred to the clean water in pot No. 1, such steam will be condensed and the transfer of its heat to the water will produce evaporation at the temperature due to a low vacuum. The vapor from No. 1, at a lower temperature than the exhaust, passes to the calandria of No. 2, and there produces a second evaporation under the influence of a better vacuum, and the vapor from No. 2 causes the evaporation in No. 3 under the influence of a high vacuum.

The condensed water drawn off by the pumps is delivered by branches L and M to a distilled water receiver in a de-aerated and perfectly pure condition.

The feed delivery from the hot well, may, and should be, filtered in order to remove grease, etc., and it may be passed through an economizer placed between the boiler and the chimney. These appliances are apart from the direct object of the illustration, and are not seen on the plan.

To enable any refrigerating engineer to estimate for a distilling apparatus of this description, proportions will be given, and the several transfers of heat be worked out for evaporators suitable for a 50-ton plant. Fifty American tons require 100,000 pounds of distilled water per twenty-four hours, and, allowing for waste 8 per cent extra, or a total of 108,000 pounds, will hardly be too much to provide for.

By following the diagram, Fig. 152, it will be seen that much more than this quantity of water can be distilled from the waste heat in the exhaust steam, even if the most economical engine in steam consumption is employed to work the compressor.

Fifty tons of ice per day, will at the very least require 115 indicated horse power; and the most economical non-condensing engines that can be made will hardly do with less than twenty pounds of steam per horse-power-hour, or 2,300

pounds per hour. At the same time the molds, apart from waste, take $\frac{100,000}{24} = 4,166$ pounds per hour.

It will be assumed, then, that in order to allow for waste, 4,500 pounds of distilled water per hour are required, and that such weight of ordinary water is supplied to the first vessel. It will also be assumed that only 1,860 pounds of the exhaust steam are to be utilized instead of the whole 2,300 pounds, such steam being under a back pressure of five pounds to the square inch, and at a temperature of 226°

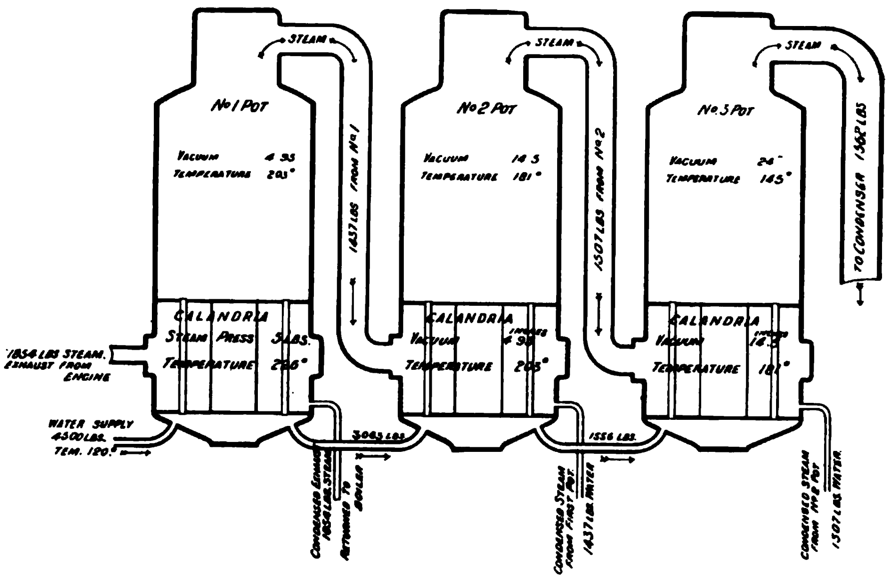


FIG. 152.—DIAGRAM OF HEAT TRANSFERS IN TRIPLE EFFECT.

The supply of water fed into the first vessel may be heated by means of coils in the chimney or flues, or by other appliances for the transfer of heat, with increase of economy. If however it be assumed that its temperature is only 120°, then the first operation will be to raise the 4,500 pounds of water from 120° to 203°, the latter being the temperature of vaporization in No. 1 vessel:—

$$203^\circ - 120^\circ = 83^\circ.$$

$$4,500 \text{ pounds} \times 83^\circ = 373,500 \text{ thermal units.}$$

Taking the latent heat of steam at five pounds gauge pressure to be 952 units, then $\frac{373,500}{952}=391$ pounds steam condensed as the equivalent of raising 4,500 pounds of water 83°.

The steam passing from No. 1 vessel is marked 1,437 pounds, therefore that weight of water has to be evaporated at a temperature of 203°, the latent heat at such temperature being 972. The latent heat of the steam in calandria is 952. Then $\frac{1,437 \times 972}{952 \text{ (latent heat)}}=1,467$ pounds as the weight of steam condensed equivalent to the evaporation.

Adding this 1,467 pounds to the 393 pounds above, gives 1,860 pounds weight of condensed water to be drawn from the first calandria, which is of course the same as the exhaust steam introduced. This water may be returned as feed to the boiler direct, or be filtered and heated in an economizer.

Deducting this weight of 1,437 pounds evaporated in the first vessel from the total of 4,500 pounds supplied to it, 4,500—1,437 = 3,063 pounds of water passing to second vessel.

This water passes in at a temperature of 203°, but as the temperature of the second vessel due to the better vacuum is only 181°, it will, in falling the difference, 203—181=22°, give off vapor as follows:—

$$\frac{3,063 \times 22}{192 \text{ (latent heat)}}=70 \text{ pounds (nearly) of evaporation.}$$

As the vapor from the top of the first vessel amounting to 1,437 pounds is condensed in the second calandria it will—being assisted by the better vacuum and lower temperature—evaporate an equal weight, or nearly so. Adding 1,437 to 70 gives a total of 1,507 pounds evaporated from the top of the second vessel.

Deducting again this weight of 1,507 pounds from 3,063 passing in at the bottom, 3,063—1,507 gives 1,556 of water to supply the third vessel. This being in direct communication with a surface condenser, and having a vacuum of twenty-four inches, the corresponding temperature will be lowered to 145°, and the water, in dropping from 181°, will part with 181°—145°=36 units per pound.

$$\text{Then } \frac{1,556 \times 36}{1,012 \text{ (latent heat)}}=55 \text{ lbs. (full) of vapor.}$$

As before, taking the evaporation in the third vessel, due to the condensation in its calandria of the vapor from the second one, to be equal in weight, or 1,507 pounds, the total will be $1,507+55=1,562$ pounds evaporated from the top of the third vessel.

The slight discrepancy between 1,556 pounds entering the third vessel and 1,562 pounds leaving—which should be of course equal—is due to slight differences in the latent heat allowed for. The sum of the different weights of vapor passing out of the three vessels to be condensed for the supply of the ice cans is $1,437+1,507+1,562=4,506$ pounds, slightly in excess of what was supplied to the first vessel. The weight of steam from the engine exhaust was 1,854 pounds, therefore $\frac{4506}{1854}=2.43$ pounds of distilled water for each pound of exhaust steam.

It will easily be understood, that by putting an exchanger on the last vessel's outlet to the condenser, where the temperature is 145°, more initial heat could be given to the water supply of 4,500 pounds weight above 120°, with improved results. If the supply is fed into No. 1 vessel at 203 then $\frac{4506}{1461}=3.08$ pounds of water per pound of steam.

The condensed vapor from the third vessel will be delivered by the main air pump from the surface condenser, and in order to take the water from the calandrias of Nos. 2 and 3, small voiding pumps or supplementary air pumps, as before described, are necessary.

Such an apparatus as that described is found in practice to require about one square foot of heating surface for six pounds of water to be evaporated per hour, therefore $\frac{4500}{6}$ $=750$ square feet, or 250 feet for each vessel.

The tubes would be about thirty inches long, one and one-half inches in diameter and No. 16 or 17 gauge in thickness.

It will be noticed in Fig. 151 that the vapor pipes differ in size. This is to make the fall of temperature between the vessels as slight as possible. The velocity of the vapor is not greater than 3,500 feet per minute into the first calandria,

4,000 feet to the second, 5,000 to the third, and 7,000 feet to the condenser.

The Colonial Sugar Refining Co., of Sydney, have numbers of these plants—some of enormous size—working at

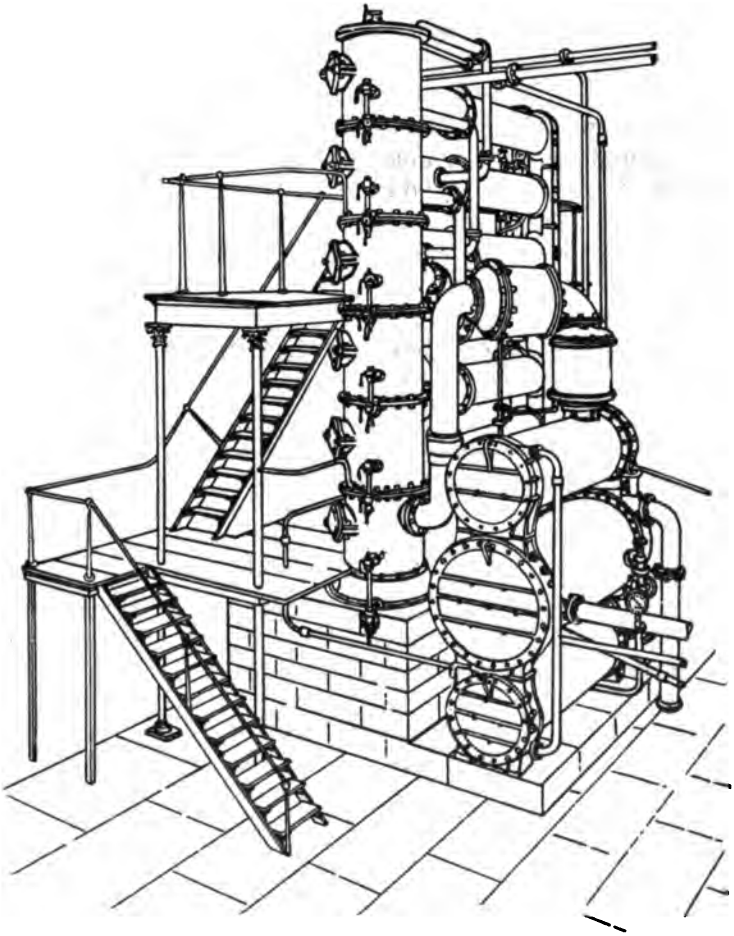


FIG. 153.—SIX-FOLD EFFECT FOR DISTILLED WATER.

their mills in New South Wales, in Queensland, and in the South Sea Islands, quadruple and quintuple as well as triple, and by successive stages they have much reduced the complication so that one-half of the cocks and fittings as used in Europe are now done away with. They have also, by the use

of large pipes giving a low velocity to the vapor, reduced the friction and loss of pressure, and largely increased the efficiency of the plant.

The author is much indebted to his friend, Mr. Hector Kidd, member Institute Mechanical Engineers, for much reliable information derived from a very wide experience with these evaporating plants, and for the information that with the company's quintuple effects as much as six pounds of water per pound of steam is evaporated, or say fifty pounds to one pound of very ordinary fuel. A paper by Mr. Kidd on this subject will be found in the third volume of the transactions of the engineering association of New South Wales.

Fig. 153 shows a sextuple effect plant suitable for such places as the dry uplands of Western Australia, where the water supply is so salt or brackish as to be unfit for potable uses. It is not so powerful or economical as the plant in Fig. 151, but is differently arranged to enable the salt deposit to be easily removed. The distilling condenser and cooler lie horizontal and communicate with the sextuple effects coupled up to the vertical column of separators.

Such machines are capable with six effects of producing four and one-half pounds of fresh water from sea water for every pound of steam raised in the boiler; and being generally independent of the exhaust steam of an engine, are worked at a much higher initial pressure and temperature than under the system shown in the larger plan No. 151.

CHAPTER XX.

SUPPLEMENTARY AND FINAL.

The loss of time necessarily involved through this work, written in Australia, being printed and published in Chicago, has sufficed for a progressive art like mechanical refrigeration to move perceptibly forward in the interval. This would seem to warrant the inclusion of the additional illustrations and remarks regarding same which follow.

A large part of the matter comprising this chapter is merely an outline of the principal distinctive features of each of the machines illustrated, with comparatively little analytical comment upon same, except in a few cases. As intimated above, the time necessary to accomplish such work would unduly advance the date of publication. It is proposed, however, to prepare for a second edition of this work an exhaustive analysis of all features of machinery and systems herein illustrated, the principles of which have not been thoroughly explained and described in this edition.

LATE TYPES AMERICAN ABSORPTION MACHINERY.

The absorption system is briefly described in Chapter VIII, and the process diagrammatically explained by Fig. 14, but no details or illustrations are there given of the various parts of the plant.

In the Vogt type of absorption machine, illustrated by Fig. 154, on following page, the noticeable feature is the absence of round coils and bent pipes throughout the entire system.

Fig. 155 shows three views of the improved generator or still, of an absorption plant, as made by the Henry Vogt Machine Co., Louisville, Ky., U. S. A., including the rectifying and analyzing devices. By the system of fractional dis-

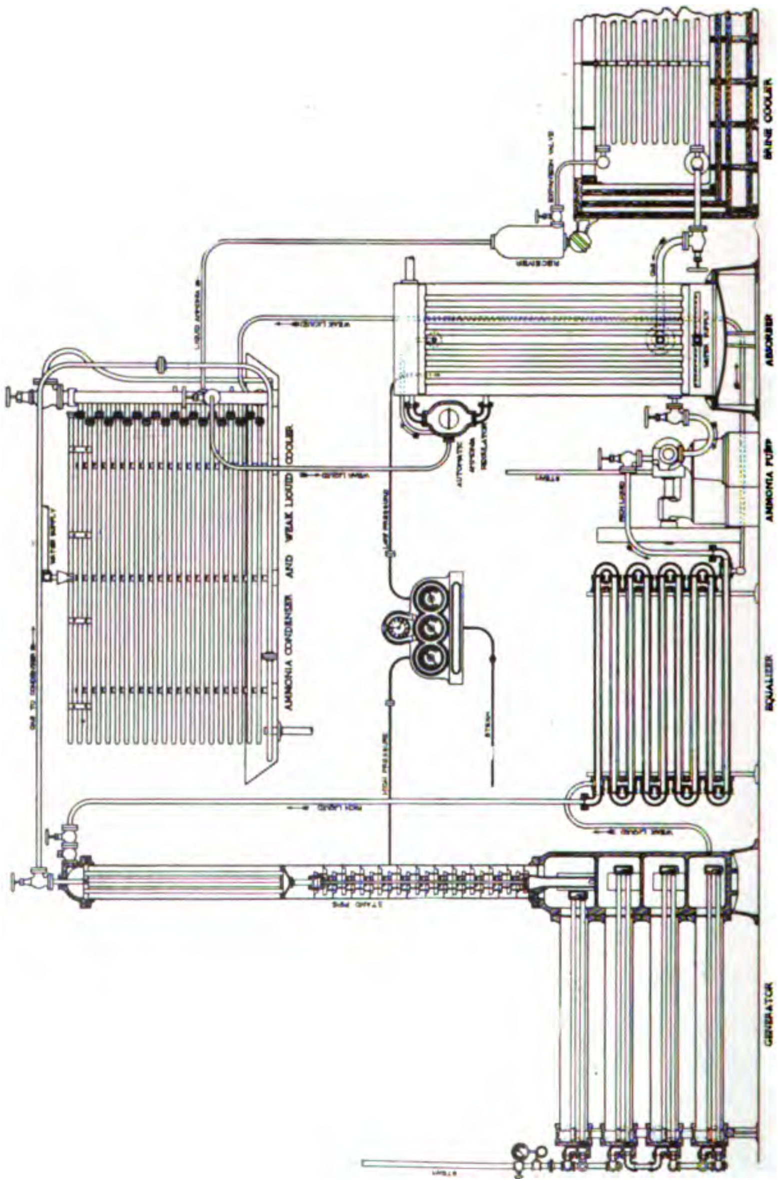


FIG. 154.—DIAGRAM REPRESENTING CYCLE OF OPERATIONS IN VOGT AMERICAN ABSORPTION SYSTEM.

tillation thus carried out, it is claimed that practically anhydrous ammonia is obtained.

The strong liquor enters by the side connection on the top of the stand pipe. The gas dissolved in such liquor is evaporated and driven off as it passes through the successive stages involved in flowing through A, B, C, D, E and F. The

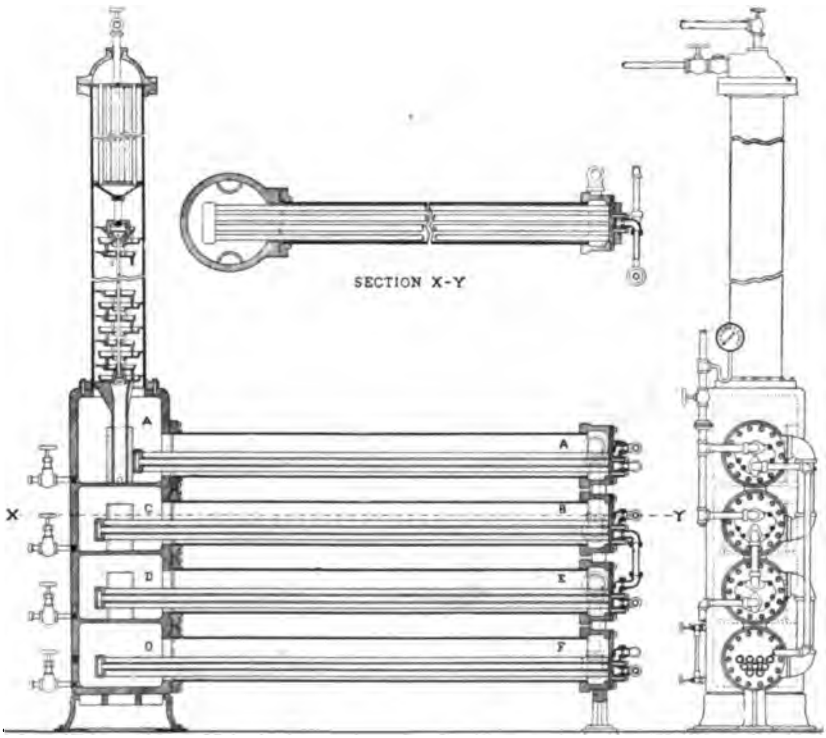


FIG. 155.—GENERATOR OR STILL FOR VOGT ABSORPTION PLANT.

liquor is left very weak by the time it reaches the compartment O.

An examination of the mechanical construction of this generator shows that it consists of a main casting, divided into four compartments, communicating with each other; and four horizontal pipes, connected to main casting, which contain the steam heating coils. The upper compart-

ment of the main casting is connected to a stand pipe containing an analyzer and rectifying coil for drying the gas before leaving the still. The strong liquor is admitted at top

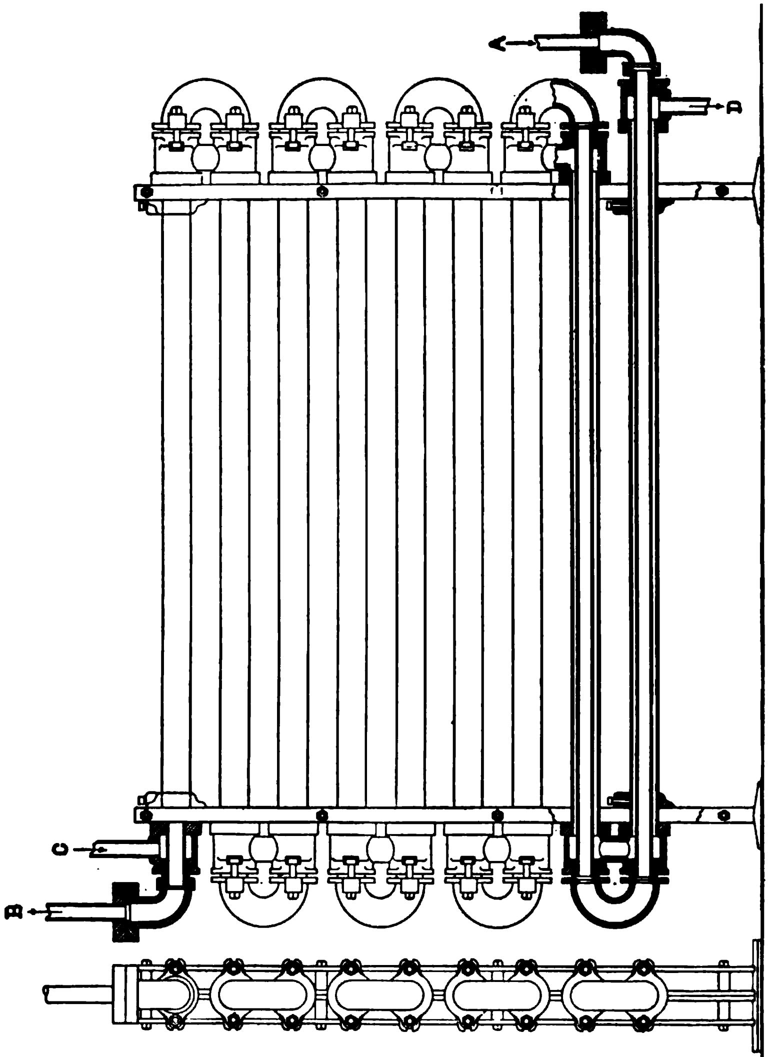


FIG. 156.—EQUALIZER OR EXCHANGER FOR VOGT ABSORPTION PLANT.

of stand pipe, passes through the rectifying coils and analyzer to the upper compartment of the main casting, flowing thence over the steam coil in the horizontal pipes from

one to the other until the lower compartment is reached. The gas generated passes through the opening in each compartment to the stand pipe, where the moisture is deposited, and the dry gas passes to the condenser.

Fig. 156 is a modern type of heat exchanger or economizer. It is made with straight concentric pipes, and is of a most mechanical and trustworthy design. It will be seen that the outer tubes are connected at the alternate ends by H pieces, and that the internal pipes are coupled by external bends, which also act as glands to the jointing. This method of construction makes what should be a thoroughly reliable job.

The strong liquor on its way to the still enters the exchanger at the bottom, leaving at the top. The weak liquor from the still enters the exchanger at the top and leaves same at the bottom.

The ammonia pump used is of the double-acting horizontal fly-wheel pattern. The special feature of this pump is the ammonia stuffing box and the water chamber surrounding it, which latter acts as a lubricator for the piston rod. The speed of the pump is twenty-five revolutions per minute.

The absorber is constructed like an upright tubular boiler open at the top. Tubes are distributed uniformly and arranged in such manner that they can be cleaned while the machine is in operation. The cooling water enters at the bottom and discharges at the top. The return gas from the expansion coils enters at the bottom and the weak liquor at the top, the flow of the latter being controlled by an automatic regulator.

The Ball American absorption machine, made by the Ice and Cold Machine Co., St. Louis, Mo., U. S. A., as originally constructed in 1878, and of five tons daily ice making capacity, was a slight modification of the Carré machine. The ice tank was eight feet square and twenty-four inches deep, and the ice cans four inches thick by eight inches wide, also eight inches square by twenty inches deep, making ice weighing twenty-five and fifty pounds each, respectively. The cans were made of galvanized iron, some of them of copper.

The original cost of building machine was $14,000.

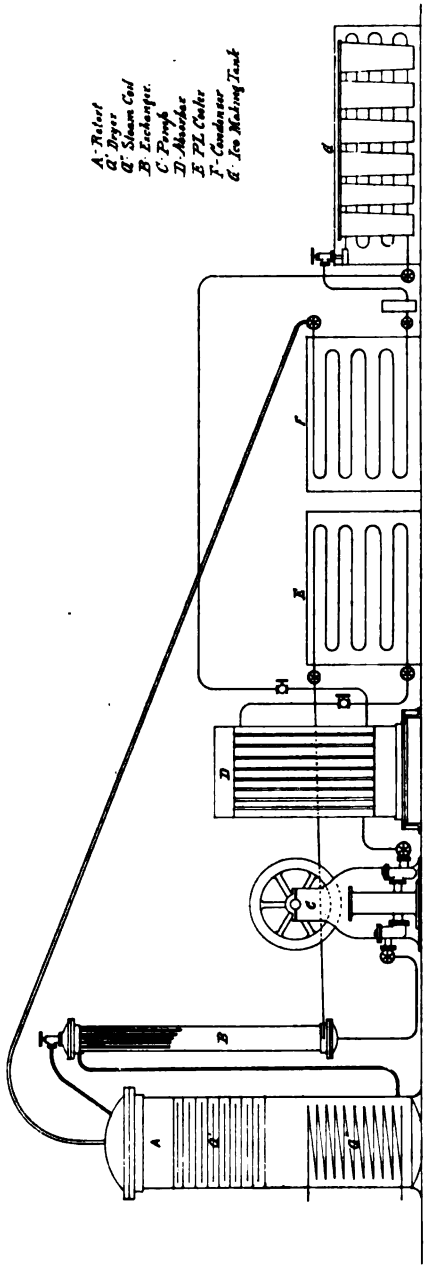


FIG. 157.—ILLUSTRATING CYCLE OF OPERATIONS IN A BALL AMERICAN ABSORPTION MACHINE. ICE AND COLD MACHINE CO., ST. LOUIS, MO., U. S. A.

After twenty-two years the machine is still of the Carré type, enlarged and made to meet the American idea of large units and expansion. See Fig. 157. The tank of eight feet square has developed into tanks 30×90 feet, and from one to four attached to one machine. Blocks of ice no longer weigh twenty-five pounds, but from 100 to 400 pounds, and, if you talked copper cans, you would be thought crazy.

The generator is a vertical cylinder of marine steel, with removable top head to same, heated with steam coil, and with drying pans in the gas dome. The condenser is of the open air or submerged type, depending upon the water. The poor liquor upon leaving the generator goes into the shell of exchanger or equalizer, which is a cylinder with removable heads containing tubes. The poor liquor, from the shell of this exchanger, goes to the poor liquor cooler coils (either of submerged or open air type), and from there to the absorber.

The gas being liquefied in the condenser goes through expansion valves to expansion coils in freezing tank, and returns from freezing tank to absorber.

The absorber is a cylindrical vessel with vertical tubes, the water passing up through the tubes, cooling the ammonia and carrying off the heat generated by absorption.

The ammonia pump consists of two single-acting vertical pumps driven by direct connected vertical engine, pumping the now enriched ammonia from absorber through the tubes of the exchanger into the top of generator, completing the cycle.

The separation of moisture in retort is exceedingly good, an air blast of 14° below zero F., being obtained under ordinary working conditions, and a temperature in the ice tank of from zero to 2° above, F., being maintained for months at a time.

In experimental machines, absorbers built of straight pipe and injecting the poor liquor into the gas returning from the expansion coils have been made with very satisfactory results, especially so where this absorber is placed from twenty-five to thirty feet above the expansion coils, and the weight of the rich liquor coming down to the rich liquor tank reducing the back pressure in the expansion coils some six or seven pounds, and allowing a very low temperature to be

carried. An absorber of this type works after the same manner as a Bulkley siphon steam condenser.

LATE AMERICAN CARBONIC ACID MACHINES.

When discussing the use of carbonic acid as a refrigerating agent or medium, the only examples illustrated were by Messrs. Hall, of Dartford, in England. Such machines are now made in Germany, in Australia, by Mephan Ferguson, of Melbourne, and in the United States.

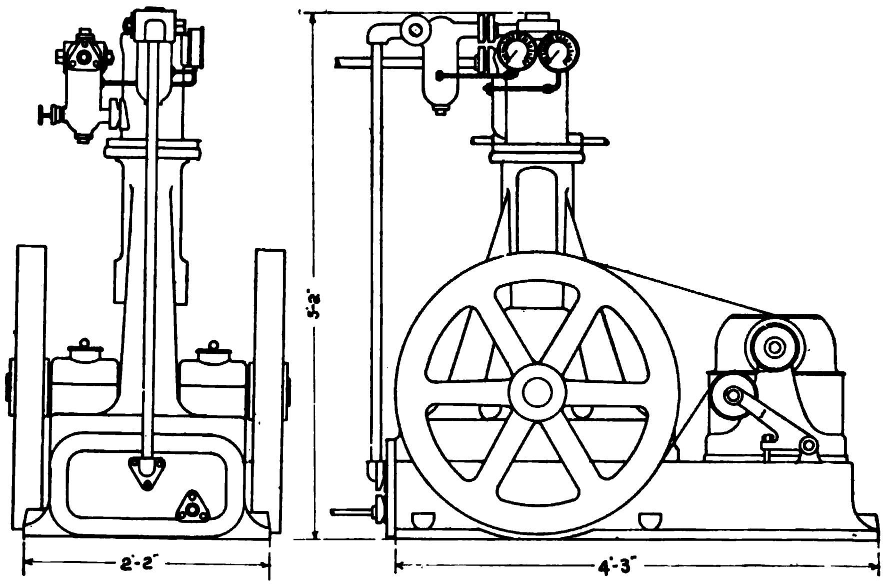


FIG. 158.—SECTIONS OF THE COCHRAN CO.'S CARBONIC ANHYDRIDE MACHINE, LORAIN, OHIO, U. S. A.

Figs. 158 and 159 are two American machines of recent introduction, which show that notwithstanding the greater power required for a given amount of refrigeration, carbonic acid has more than compensating advantages for small units, and in special circumstances. This is owing to its innocuous character, and the absence of danger in the case of an escape of gas from the machine.

Messrs. Kroeschell Bros.' machine has an extremely neat and mechanical looking appearance. That by the Cochran Co. is made more complete and portable by having the con-

denser combined on one sole plate with it. The illustration, Fig. 158, shows a cross and transverse section of their simple motor driven compressor, and is one of their latest designed machines.

SOME LATE AMERICAN CONDENSERS.

Fig. 160 is an atmospheric condenser for an absorption plant, made by the Henry Vogt Machine Co., Louisville, Ky., U. S. A., and is of the type (described on page 82 *ante*) where vertical headers are connected by zigzag coils laid horizontally. These zigzag coils form a two-storied condenser, as they are built in two separate sections. The gas condenser

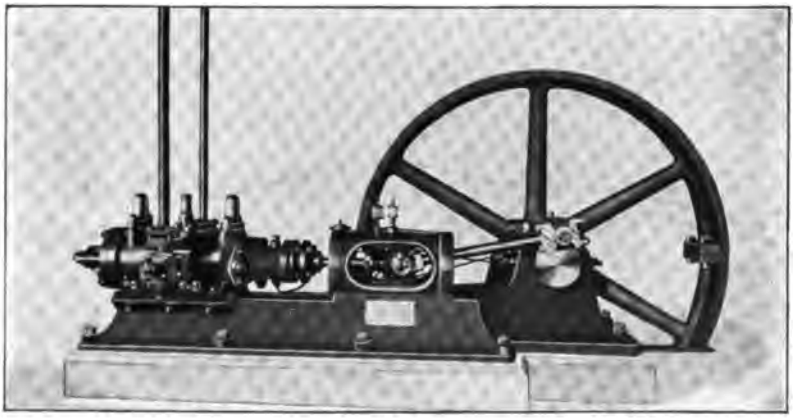


FIG. 159.—KROESCHELL BROS. CARBONIC ACID MACHINE. CHICAGO, ILL., U. S. A.

headers are set over those for the weak liquor, and this saves condensing water, as the weak liquor on its way to the absorber is cooled by the waste water from the condenser proper. Besides these two exchangers forecooler coils are shown in the water tray below, and the arrangement is such that the hot gas first enters these submerged coils, where it is partially cooled by the water from the coils above. The gas then passes to the top of the first condenser headers, flowing horizontally through the coils until the anhydrous liquor is drawn off at B.

From D to C is the weak liquor cooler, the liquid entering at D and flowing upward—in the reverse direction to the

cooling water—to its exit at C. Practically this is a three-story or triplex condenser, and it is as such (and as an illus-

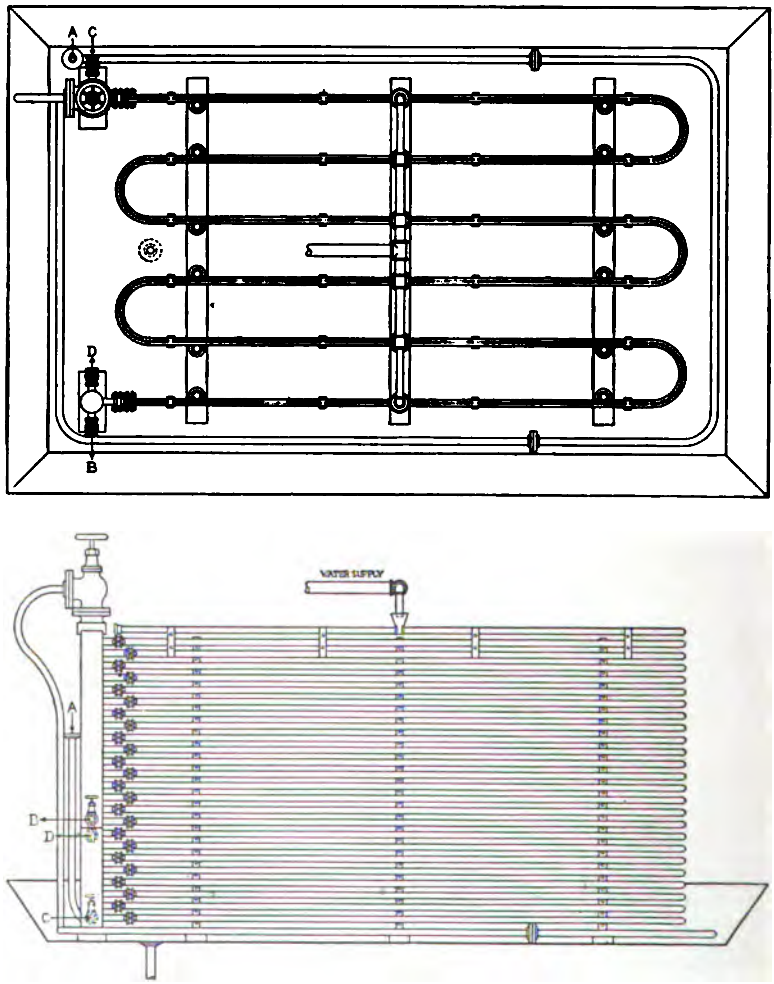


FIG. 160.—TRIPLEX AMMONIA CONDENSER FOR VOGT AMERICAN ABSORPTION MACHINE.

tration of a modern detail to secure economy in the use of condensing water) that it is introduced.

Submerged condensers are used with this machine, especially where the water is impure and contains much lime.

Fig. 161 is an illustration of an ammonia condenser of the Ball type. It is especially effective where the cooling water is warm or scarce, and was originally constructed in 1890. Since then it has been extensively copied by other builders.

As seen, the hot gas from compressor is admitted in the lower pipes and after having the sensible heat taken out of same it is piped to the top of condenser where the fresh water comes in contact with the cool gas, instead of being heated by hot gas, as in the old type of condensers formerly constructed. The saving in actual expense is claimed to be about

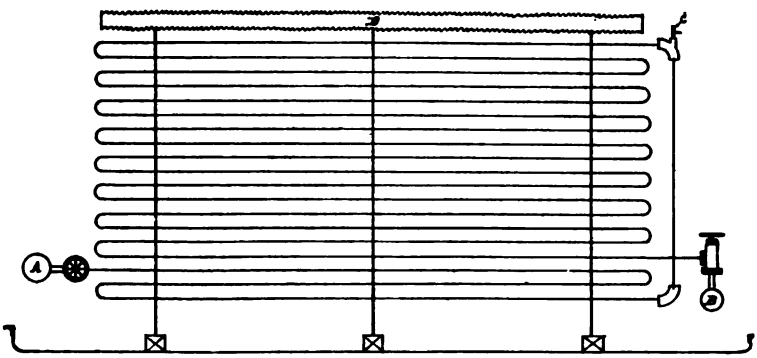


FIG. 161.—BALL AMMONIA CONDENSER.
A.—Hot Gas Header. *B.*—Liquid Header. *C.*—Air Valve. *D.*—Sprinkling Trough.

15 per cent in the water bill, or about 10 per cent in the condenser pressure. Condensing liquid has the same direction of flow as the gas, and not an opposite flow, as in some condensers.

Fig. 162 is a section of Frick Co.'s latest design of atmospheric ammonia condenser. In the construction of this condenser the manufacturers have aimed to have the cold water come in contact with the coldest gas, which, becoming warmer as it meets the warmest gas, flows off over the hot gas pipes from compressors, finally passing off through the overflow.

The ammonia condensers designed and constructed by the Fred W. Wolf Co., Chicago, are of the atmospheric type

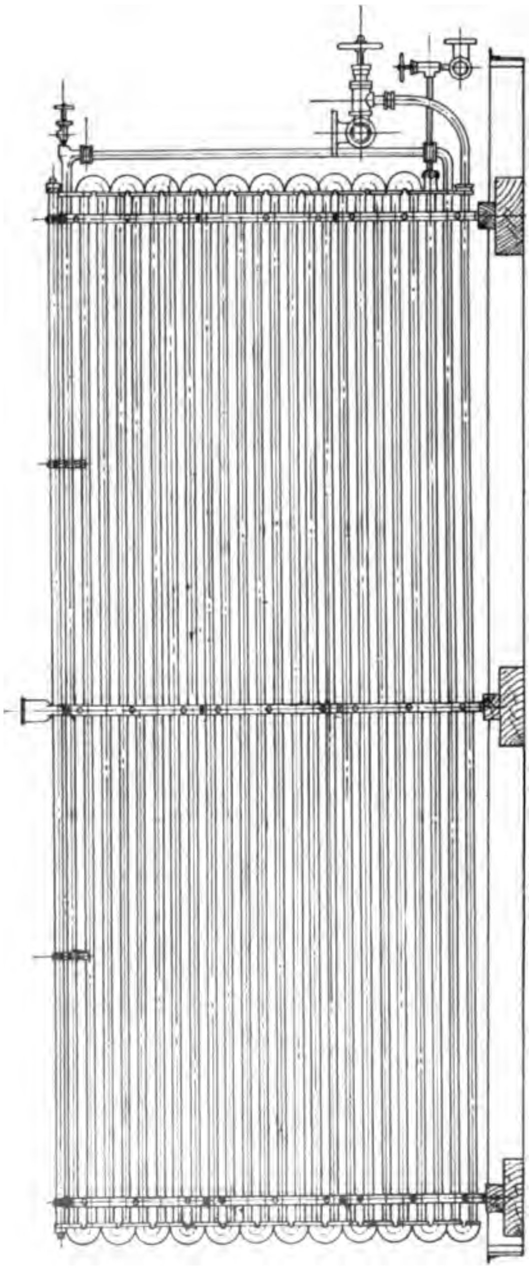


FIG. 162.—FRICK CO.'S ATMOSPHERIC AMMONIA CONDENSER.

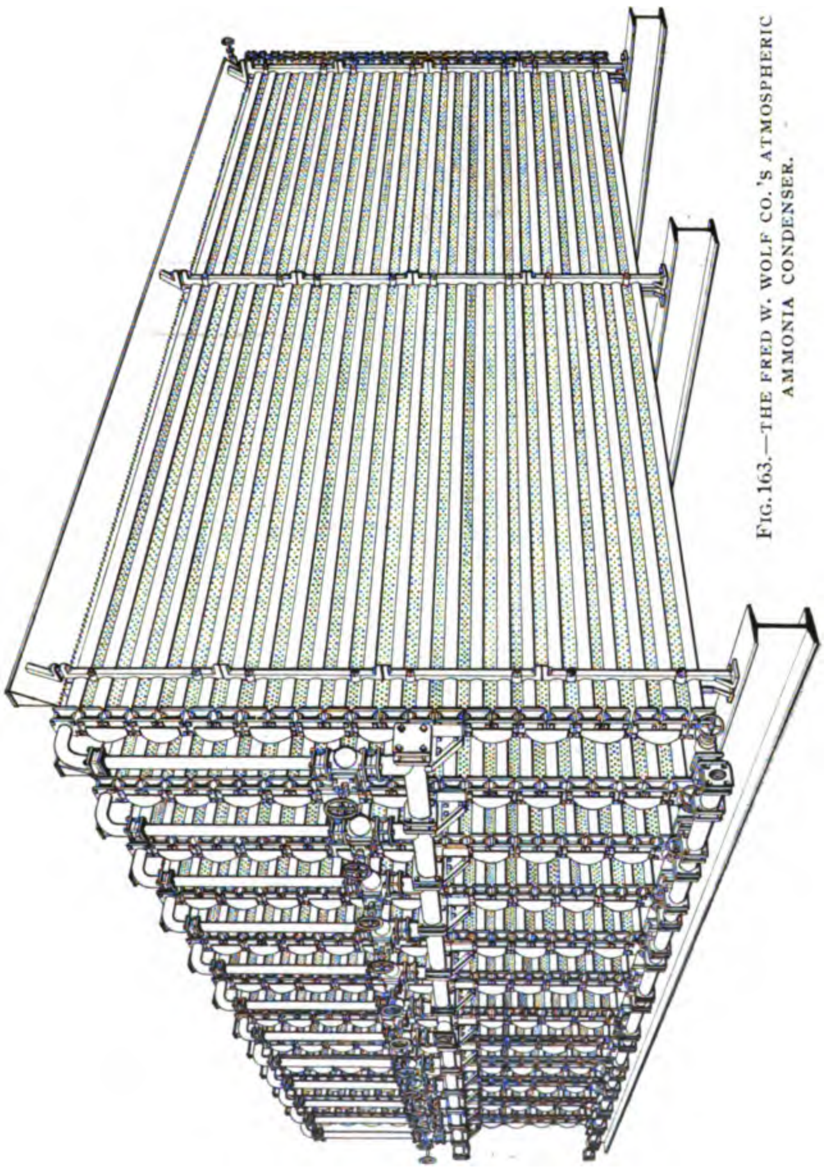


FIG. 163.—THE FRED W. WOLF CO.'S ATMOSPHERIC AMMONIA CONDENSER.

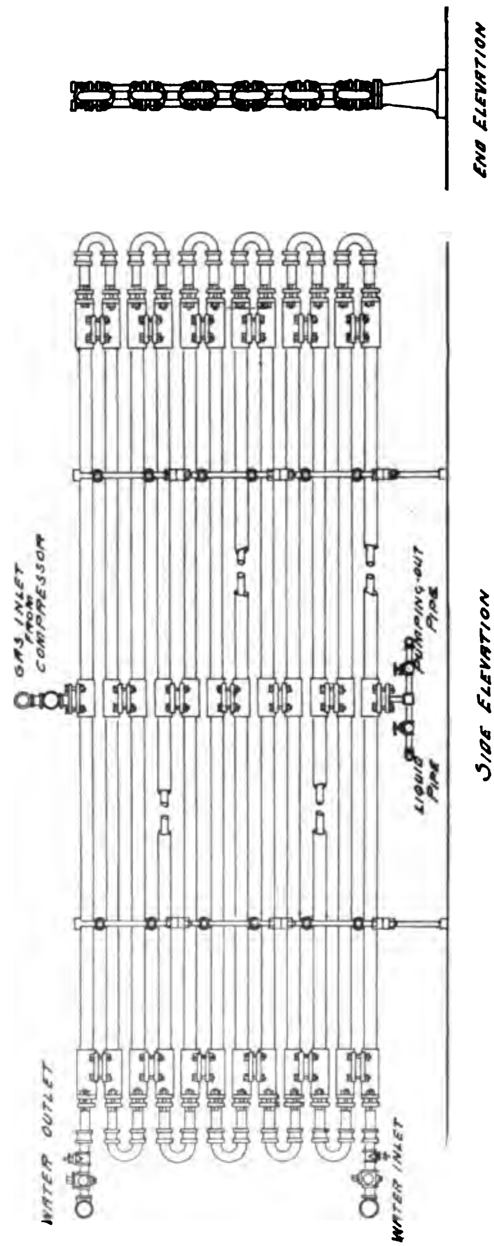


FIG. 164.—WESTERLIN & CAMPBELL'S PATENT DOUBLE-PIPE CONDENSER, CHICAGO, ILL., U. S. A.

(see Fig. 163), the standard size of each section being twenty-four 2-inch pipes twenty feet long. These pipes are manufactured from selected skelp, and the drop forge Bessemer steel flanges are screwed on to same while hot, thereby allowing the flange to shrink on as it cools. These condensers are supplied with galvanized iron water troughs with patent leveling device, and between the pipes is fastened a perforated steel strip, thereby allowing a free circulation of air. Each section of these condensers is supplied with an inlet and outlet valve, thereby allowing each section to be evacuated of ammonia without interfering with the operation of the remaining sections when connected with the suction of the machine.

Fig. 164 shows a side and end elevation of the Westerlin & Campbell patent double-pipe ammonia condenser, an invention of recent date, constructed with a view of incorporating all of the best and most practical features of both the submerged and atmospheric types of ammonia condensers. Reference to the cut will show that the condenser is constructed with a small pipe encased within a larger pipe; usually the internal pipe is one and one-fourth inches and the external pipe two inches. The gas inlet is located in the center of the coil, and the hot ammonia gas enters the space between the 1¼-inch and 2-inch pipes, spreading both ways from the center toward the two ends. At the ends the gas travels down to the next space between the pipes below, where it travels from both ends toward the center, and again spreads toward the two ends in the next succeeding space, the object being to film the gas out to the greatest possible extent, bringing all of the gas in direct contact with both the internal and the external cooling surfaces. The water enters the 1¼-inch pipe at the foot of the condenser and travels back and forth upward until it overflows into the manifold at the top and end of the condenser. It will be noticed that while the travel of the gas is downward the travel of the water is upward, making an interchange of temperature that results in the warmest water meeting with the current of the warmest gas, and the gas is gradually cooled down and condensed into liquid as it travels along, meeting with the cooler water, until finally the ammonia liquid is discharged at

the bottom of the condenser at a temperature as low as the temperature of the initial water in the internal pipe. At the foot of the condenser connections are provided for conveying the liquid ammonia to the liquid receiver, and also for connecting to the suction pipe of the machine, or to the absorber, in case the condenser is used in connection with an absorption machine, so that the gas inlet and liquid outlet can be closed and all of the gas in the condenser can be drawn out in case of necessity for repairs without interfering with the operation of the balance of the plant. The condensers are usually erected in nests of several stands, and it is always possible to cut out one stand for repairs without shutting down the plant. The water connections are cross-connected in such a manner that the water current can be reversed when it is desired to wash out the internal pipe. It has been asserted by engineers that such a construction of condenser could not be used in connection with waters badly impregnated with scale forming properties, but a considerable experience with the worst waters in America has demonstrated that the scale will not form in the pipes at all, even after more than a year of continuous operation, the scouring of the rapid current of water through the internal pipe positively preventing deposit of scale, sand or mud. No water is used over the outside of the condenser, consequently it can be located at any desired point about the plant, without necessity for water pans or tight floors. The condenser can also be placed at any desired level, as the water can be delivered to any height above the coils.

COOLING TOWERS.

On pages 64 to 67, *ante*, reference is made to the re-use of condensing water and the different arrangements by which this may be effected. Fig. 24 shows an evaporative condenser in a cooling tower. Recently several new devices for accomplishing the same result have been devised, notably that embodied in the patents of John Stocker, St. Louis, Mo., U. S. A., which shows a high degree of efficiency. A very ingenious method for distributing the water over the tower is adopted, the cooling surfaces being so arranged that a perfectly even discharge of air over the water is accomplished by means of two fans instead of one.

SOME RECENT AMERICAN VALVES.

Fig. 165 is a section of the Ball valve used by the Ice and Cold Machine Co., of St. Louis, Mo. Its construction is re-

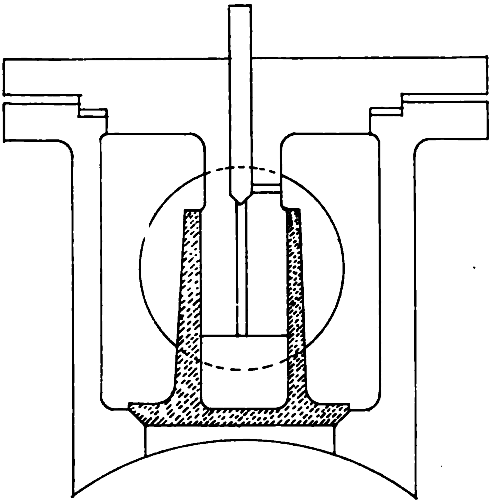


FIG. 165.—BALL DISCHARGE VALVE.

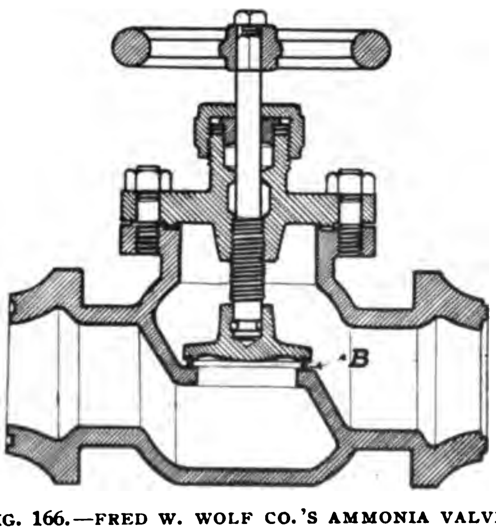


FIG. 166.—FRED W. WOLF CO.'S AMMONIA VALVE.

ferred to in the description of the Ball compression machine found on page 303.

The ammonia globe valves, manufactured by the Fred W. Wolf Co. (see Fig. 166), each contain the soft metal seat at B. These valves are well proportioned, of good weight, and are made to stand a pressure of 500 pounds; further-

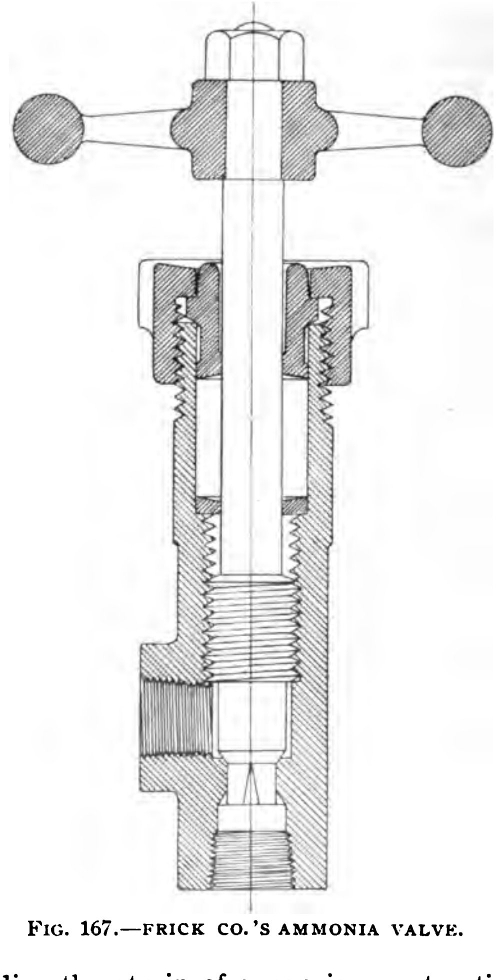


FIG. 167.—FRICK CO.'S AMMONIA VALVE.

more, standing the strain of expansion, contraction and the weight of pipe and settling. They are also so constructed that they need not be closed to repack stuffing boxes, as in having the valve entirely open any leak through the stuffing box is entirely obviated.

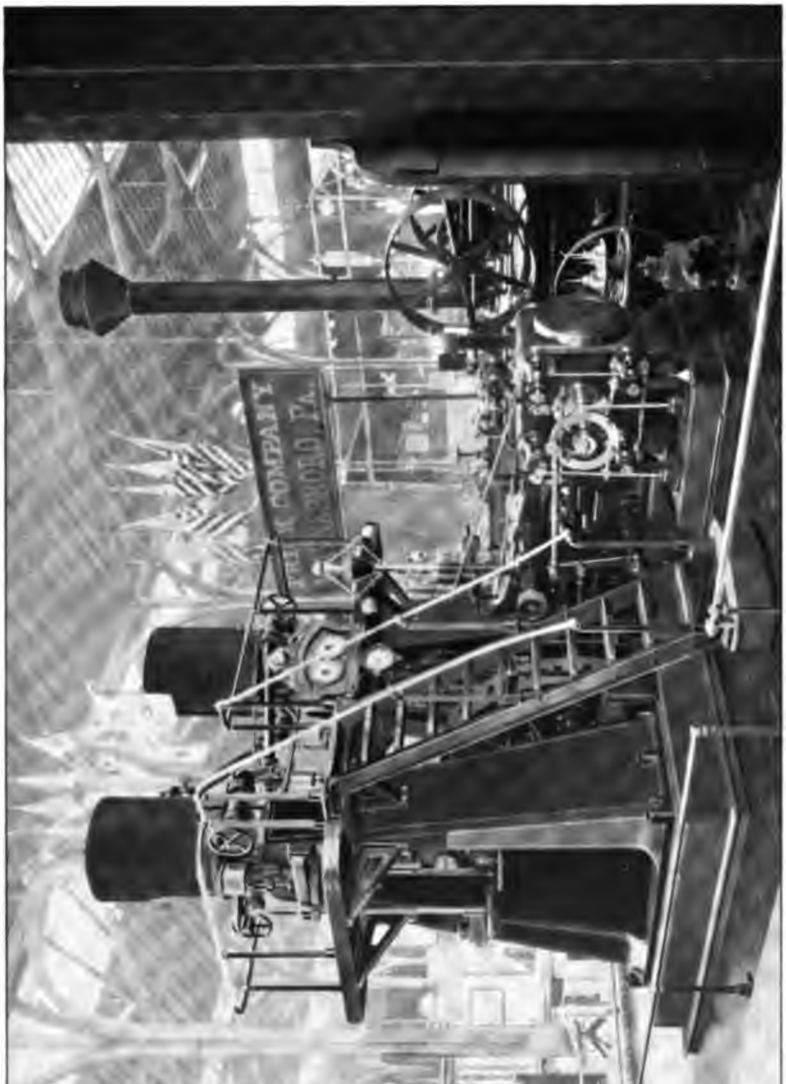


FIG 168.—FRICK CO.'S EXHIBIT OF ICE MACHINERY AT THE NATIONAL EXPORT EXPOSITION, PHILADELPHIA, 1899.

The latest construction of the Frick ammonia valve, referred to in Chapter XIII of this work, is shown in section by Fig. 167, on page 278.

THE LATEST DESIGNS OF AMERICAN AMMONIA COMPRESSION MACHINERY.

Fig. 168 is a half-tone illustration of Frick Co.'s exhibit at the National Export Exhibition, Philadelphia, 1899.

FIG. 169.—ELEVATION FRICK CO.'S LATEST AMMONIA COMPRESSOR CYLINDER, WAYNESBORO, PA., U. S. A.

Figs. 169 and 170 are the elevation and section respectively, of the most recent pattern of "Eclipse" pump—in other words, the ammonia compressor cylinder—as now made by the Frick Co., of Waynesboro, Pa.

This new design is worth careful study by both the student and hard-shell engineer, as it is a good example of

how efficiency may be combined with simplicity. It will also be noticed that it embodies those special qualities upon which

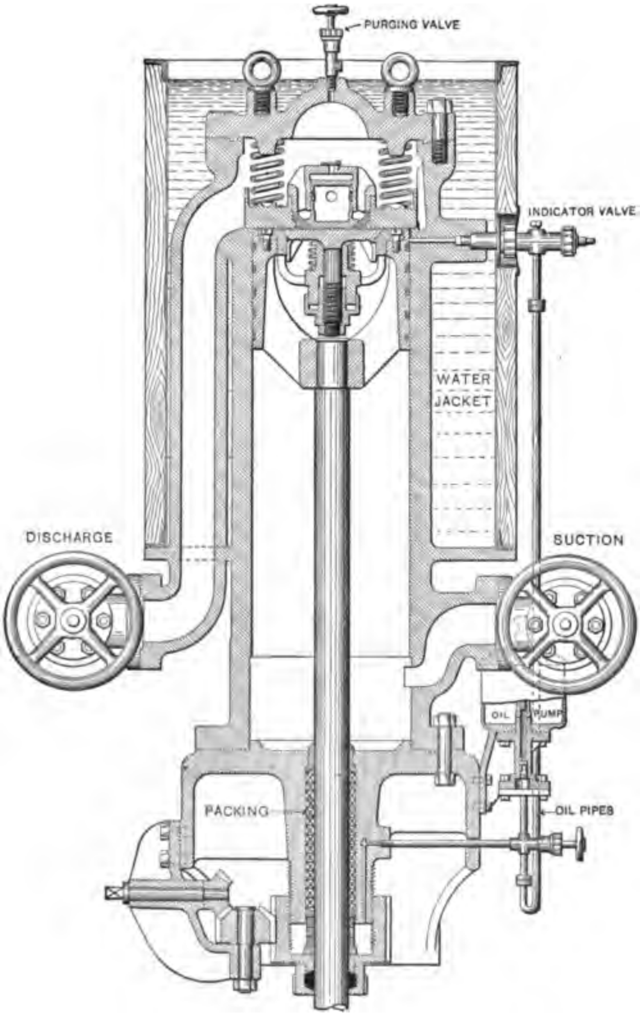


FIG. 170.—SECTION FRICK CO.'S LATEST AMMONIA COMPRESSOR CYLINDER, WAYNESBORO, PA., U. S. A.

so much stress was laid (as being desirable in such cylinders) on pages 110, 111, 123 and 151, *ante*. It was there argued that plain barrel cylinders, without attached feet or encir-

cling passages, favored homogeneous castings of sound and solid metal; and this is effected in the pump under notice, by the simple but elegant device of detaching the delivery passage from the main body of the casting, for the whole length of the piston's travel in the cylinder. The connection of the pipes to the inlet and outlet branches is simplified, by bringing them both below the water jacket; this leaves the

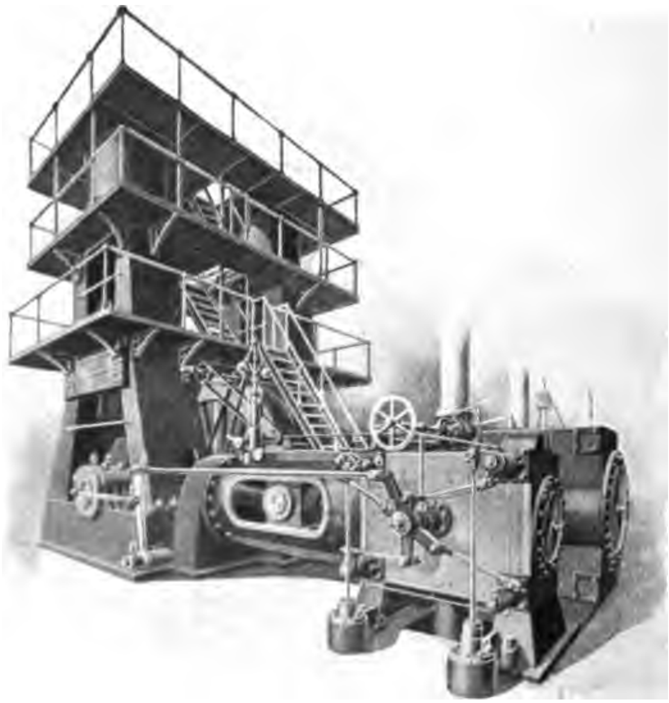


FIG. 171.—YORK MFG. CO.'S MAMMOTH MACHINE, 400 TONS REFRIGERATING CAPACITY.

head quite clear, and makes inspection of the valves an easy matter.

No lantern bushes are shown in the piston rod packing, but the stuffing box is still longer than many engineers consider necessary or even desirable; and oil is fed below the packing. This pump should be compared with that shown by Fig. 58, page 102; both aim high, but seek perfection in

different ways, and both are better than the one illustrated by Fig. 64, to which, nevertheless, the indebtedness of Fig. 58 for some ideas is gratefully acknowledged.

The York Co., of York, Pa., have only so far been represented in this work as manufacturers of compound ammonia compressors, and by Figs. 66 and 70. In Fig. 171

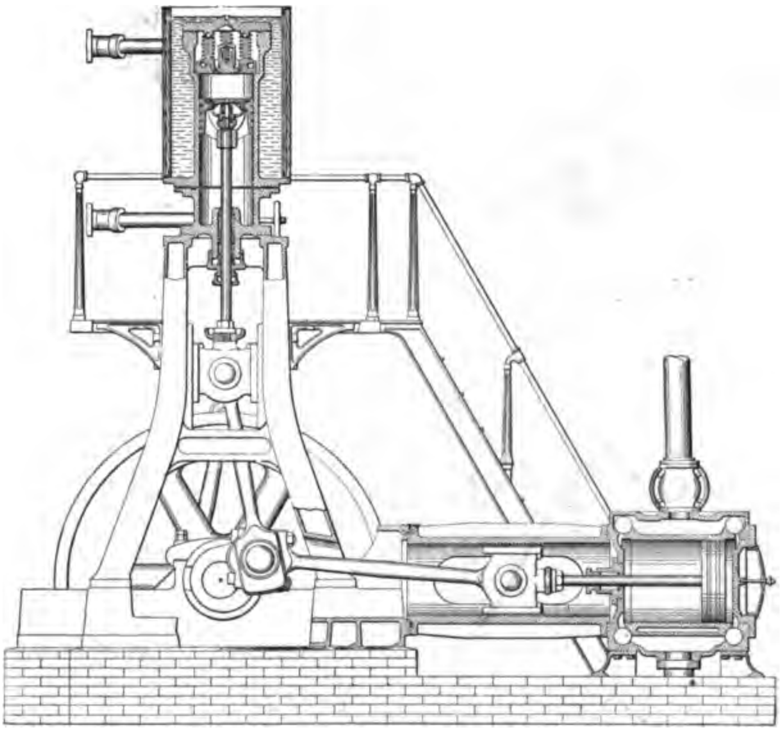


FIG. 172.—SECTION YORK MFG. CO.'S COMPRESSOR.

there is a perspective view of a modern mammoth machine made by the same builders, and equal to 400 tons refrigeration. It has two single-acting compressors, thirty inches diameter, forty-eight-inch stroke, fitted with cross-compound condensing steam engine; high pressure cylinder, thirty-inch bore; low pressure cylinder, fifty-eight-inch bore, forty-eight-inch stroke. The crank shaft has two throws and four bearings. The machine is fitted with one fly-wheel in the center

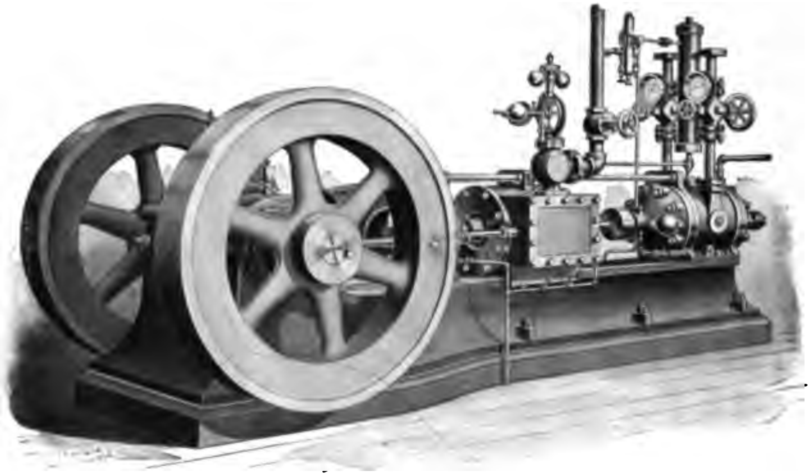


FIG. 173.—PENNEY'S HORIZONTAL DOUBLE-ACTING COMPRESSOR, NEWBURGH ICE MACHINE AND ENGINE CO., NEWBURGH, N. Y., U. S. A.

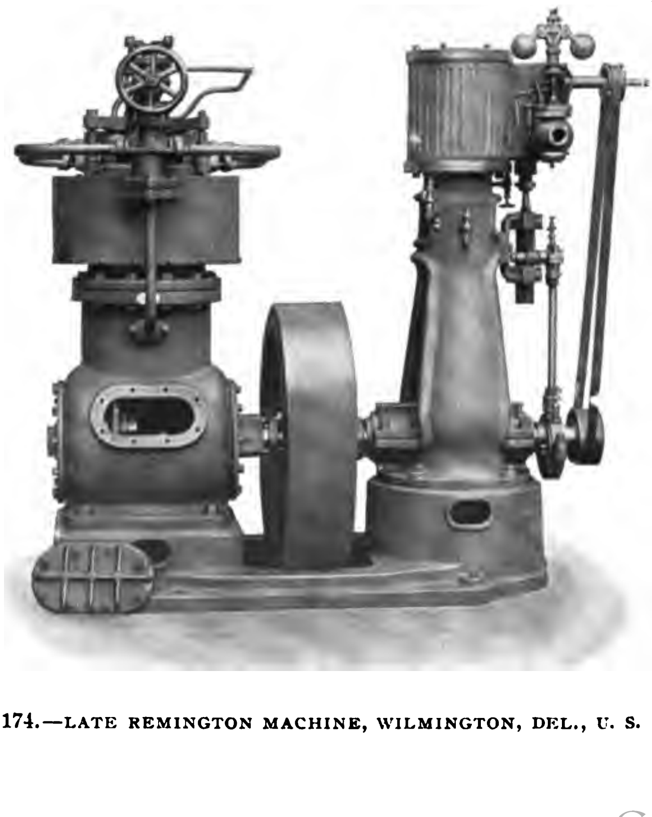


FIG. 174.—LATE REMINGTON MACHINE, WILMINGTON, DEL., U. S. A.

of the bed plate, between the two cranks. The weight of this machine when completed was 400,000 pounds.

Fig. 172 shows a section of the York vertical machine, late design.

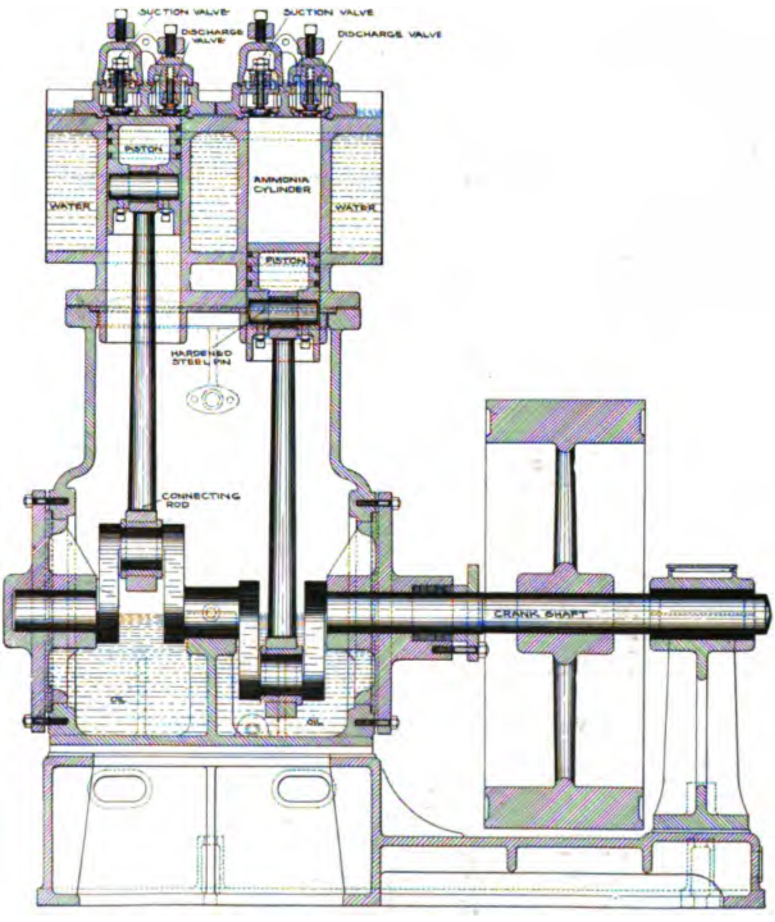


FIG. 175.—SECTION OF REMINGTON MACHINE, WILMINGTON, DEL., U. S. A.

For the reasons given on pages 119 and 120, straight-line ammonia compressors have not been greatly favored in the past, although it is common enough for compressed air. In Fig. 173, the Penney machine, made by the Newburgh Ice Machine and Engine Co., Newburgh, N. Y., U. S. A., is a

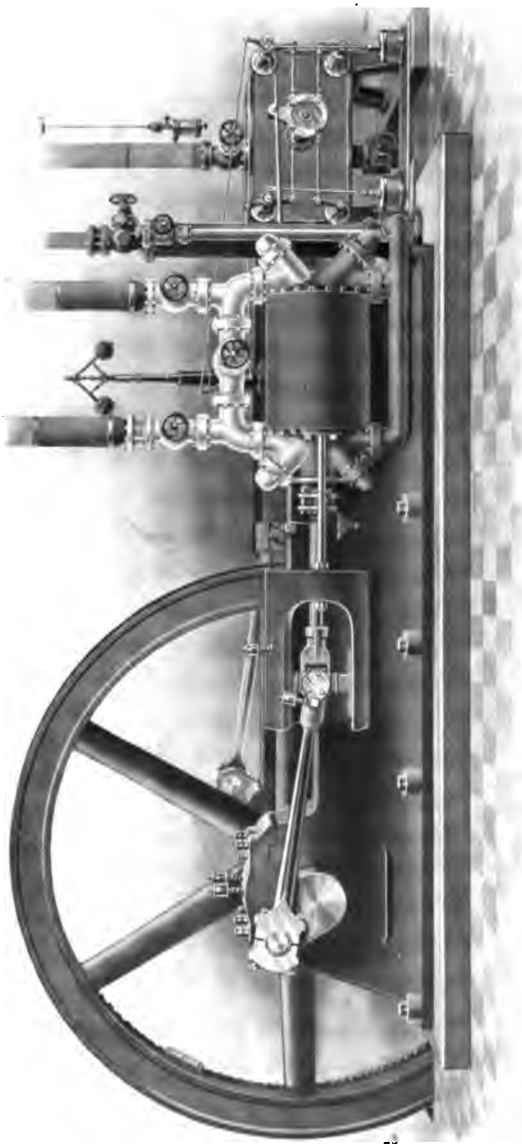


FIG. 176.—AMERICAN TYPE LINDE MACHINE, STANDARD FRAME, FRED W. WOLF CO., CHICAGO, ILL., U. S. A.

FIG. 177.—AMERICAN TYPE LINDE MACHINE, TANGYE FRAME, FRED W. WOLF CO., CHICAGO, ILL., U. S. A.

modern example of this type, and the heavy character of the fly-wheels, supporting what is said on pages 123 and 124, is very clearly apparent.

The Remington vertical compressor, as shown in Figs. 174 and 175, is of the single-acting, inclosed crank type, and has but one stuffing box, that on the revolving shaft. The ordinary type of trunk piston is used, and the crank shaft is supplied with a center bearing in order to provide for a rigid construction, and at all times runs in oil.

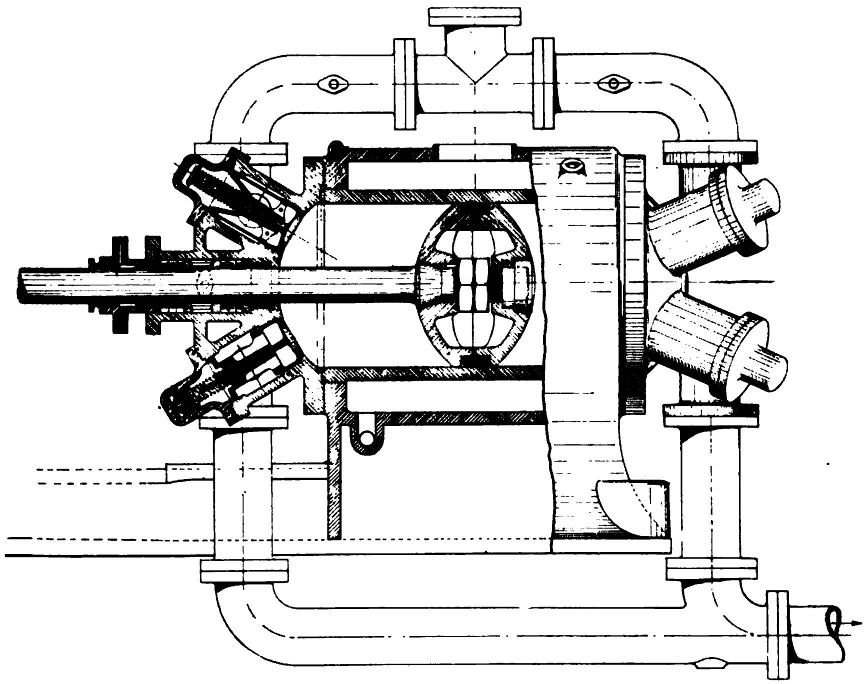


FIG. 178.—SECTION AMERICAN LINDE COMPRESSOR CYLINDER, FRED W. WOLF CO., CHICAGO, U. S. A.

There are two cylinders made in one casting provided with heads in which are located the suction and discharge cages and valves. These cages and valves are readily accessible by removing the cross-bars on top of the heads, without breaking any other joints than those directly over the cage.

The heads of the two cylinders are connected on the suction side to a common strainer box for catching the dirt and

sediment, and the discharge side to a throttle valve common to both cylinders.

The American type of the Linde machine, as manufactured by the Fred W. Wolf Co., Chicago, shows many variations in construction from the original Linde machine, as designed by Prof. Carl Linde, in 1875, the construction and operation of which have been thoroughly described and explained in Chapter XV of this work. Illustrations of the latest type American Linde machine are inserted here to show these variations in construction.

These machines are of the ammonia compression type, operating on the humid gas system.

Figs. 176 and 177 represent the "Standard" and "Tangye" styles of frames, the working parts of each, however, being of the same general design and construction.

Fig. 178 is a sectional view of the latest Wolf design of the Linde compressor cylinder.

The refrigerating machine as built by the Vilter Manufacturing Co., Milwaukee, Wis., U. S. A., illustrated by Figs. 179 and 180, consists of one or two horizontal double-acting ammonia compressors driven by one horizontal engine, generally of the Corliss type, built also by the same firm. The engine and compressor cranks are keyed on the ends of the shaft at angles to each other, bringing the highest gas pressure in the compressor at a point where the engine gets the highest steam pressure.

The ammonia compressor, as shown partly in section and partly in perspective, is generally cast with slides and pillow block in one piece. After the guides and frame—that is, the water jacket of the compressor—are bored, a cylindrical bushing is forced into the water jacket, forming the compressor wearing surface proper, and then a finishing cut is taken in one setting of the entire frame through the guides and compressor, for the purpose of making the guides absolutely true with the compressor.

The four ammonia compressor valves are placed in the two circular heads. The heads fit into a recess, and are packed with a metallic packing.

The suction and discharge valves are readily accessible, which, together with the stems, are made of forged steel, and

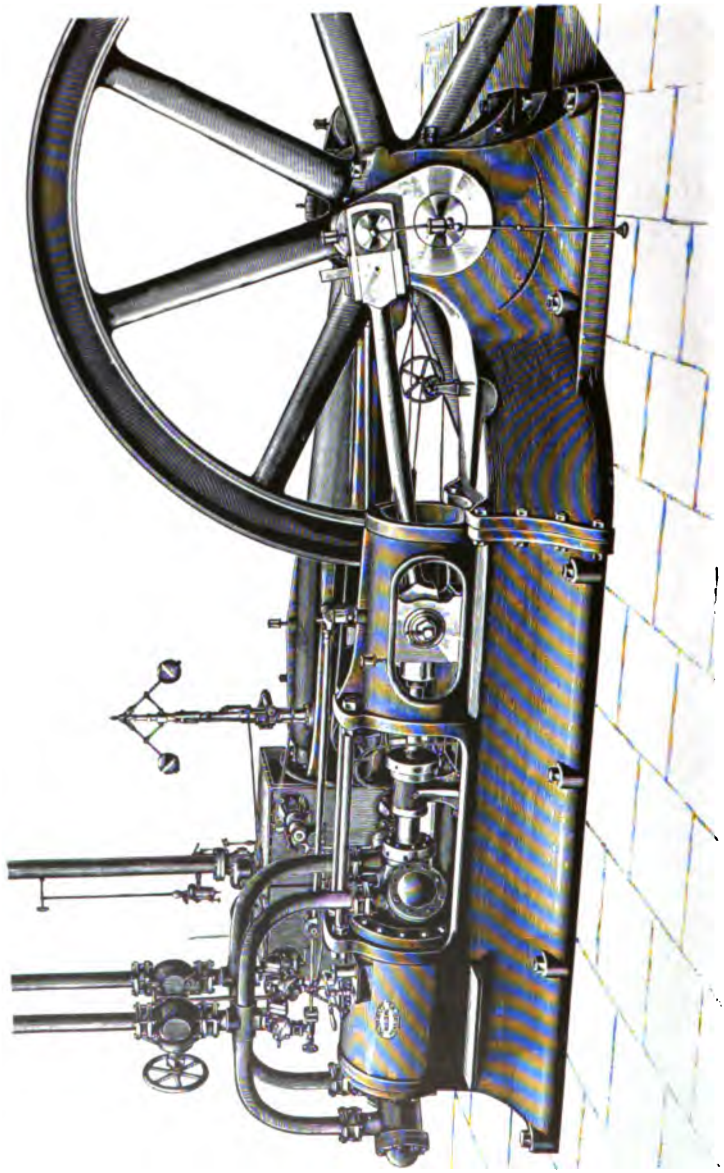


FIG. 179.—THE VILTER TYPE OF HORIZONTAL, DOUBLE-ACTING AMMONIA COMPRESSOR, VILTER MFG. CO., MILWAUKEE, WIS., U. S. A.

are provided with gas cushions, to avoid crystallization and noise in the working of the valve. The valve seats are of cast steel, turned true, and fit with a ground joint in the compressor head, making the use of packing unnecessary.

The compressor plunger is provided with self-adjusting packing rings having bull rings besides, which can be replaced easily. The piston and follower are turned to a circle to fit exactly into the front and back heads respectively of the compressor. The clearance between the plunger and the head is thereby reduced to a minimum. The length of the plunger rod can be adjusted, so as to divide the clearance equally at both ends, and so take up the wear of the crank and cross-head boxes.

The stuffing box consists of a metallic packing in the head, which is held in position by a long hollow sleeve, through which oil is circulated by means of an automatic oil pump, and this oil is used for lubrication of the plunger rod, as well as for forming a seal against the escape of ammonia. The outer end of this hollow sleeve is held in position by a separate support, which is bolted to the compressor frame proper, and at the outer end of this support a packing is provided for retaining the oil. By this arrangement of the stuffing box, it is claimed by the manufacturers that they are enabled to operate the compressor with a very much higher discharge pressure than could otherwise be effected.

Proper by-pass, or cross-connections, are placed between the suction and discharge pipes close to the compressor, so that the valves can be operated for pumping out the condenser without leaving the engine room.

The cross-heads are provided with adjustable shoes by wedge adjustment; the connecting rods have solid heads, and the crank pin is provided with a brass box lined with babbit metal, the cross-head box being of solid brass. The wear of both boxes is taken up by wedge adjustment.

If two ammonia compressors are driven by one engine, they are generally arranged tandem, and both connecting rods are coupled to one crank pin. Larger compressors are often driven by compound non-condensing or compound condensing engines, and if there is a scarcity of water, cooling towers may be added, so that water can be cooled and re-used.

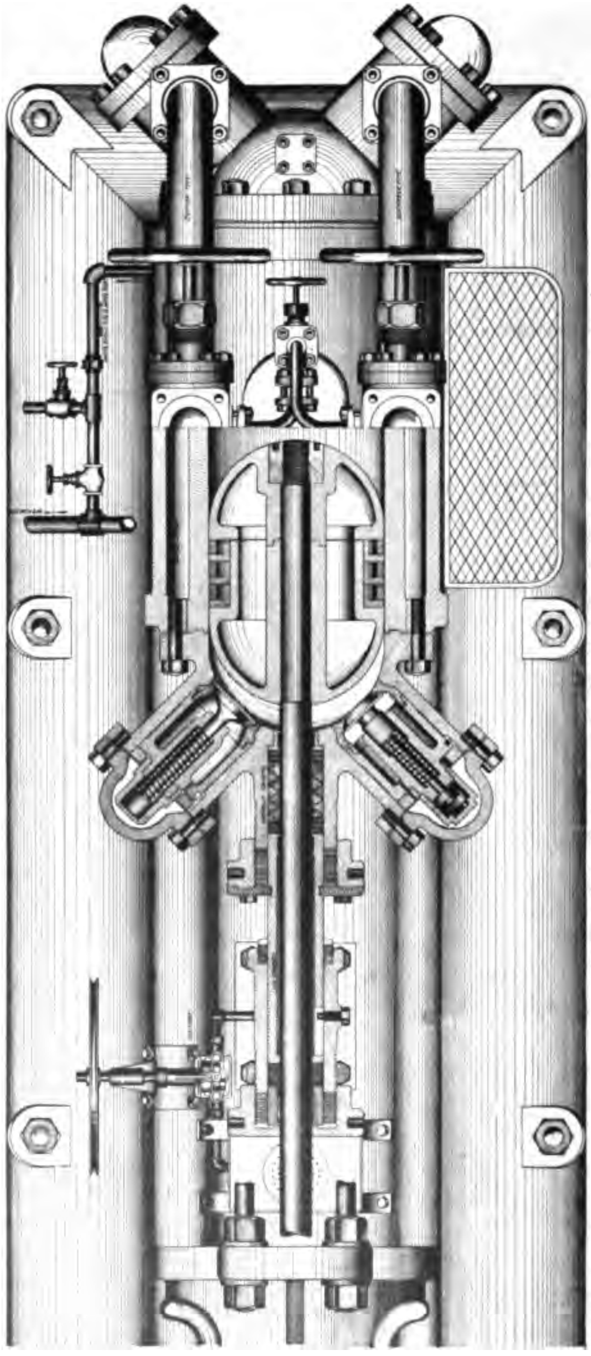


FIG. 180.—SECTIONAL VIEW VILTER TYPE OF HORIZONTAL, DOUBLE-ACTING AMMONIA COMPRESSOR, VILTER MFG. CO., MILWAUKEE, WIS., U. S. A.

The compressors may also be driven by belt or rope transmission from a line shaft operated by an engine, also doing other work, or by electric motor, or by water power.

Fig. 181 shows a perspective view and Fig. 182 a section of the compressor of the Triumph Ice Machine Co., of Cincinnati, Ohio, U. S. A. This compressor is of the horizontal double-acting type, and is fitted with five valves, three suction and two discharge. The third, or auxiliary suction valve, is perfectly balanced, and is much lighter than the main suction valves. The main suction valves must of necessity be of sufficient size to admit the charge of gas quickly at the beginning of each stroke. The springs controlling them must therefore have an appreciable tension, and it can be readily seen that in consequence the pressure of the gas in the cylinder during admission is less in the suction pipe by just the tension of these springs.

The construction of the suction valves is as follows: A guard is screwed on to the stem, fitted inside of the cage, and is ribbed so as to reduce the port area, the stem being made larger at the bottom for this purpose. Both suction and discharge valves have a stem leading to them through the stuffing box, and can be handled from the outside, thereby allowing any tension to be brought on the springs at any time. This, it is claimed, is necessary on account of machines being worked at different pressures and their relative temperatures. For instance, one side of a machine may be called upon to work on a temperature of 10° to 15° below zero, the other side to 10° or 15° above, consequently the springs on one of them would have to be changed so as to make it operate properly, and also enable the engineer to know by observation whether they are opening and closing at the proper time, and whether they have the proper amount of lift, etc.

The stuffing box has three compartments for packing, and is fitted with a relief valve, which leads into the suction. The piston is shrunk onto the piston rod, making it a perfect fit. The heads are concave, and of such a radius as to obtain a larger valve area.

Every part of the compressor is accessible. The main shut-off valves are so constructed that they can be packed while the machine is in operation.

FIG. 181.—TRIUMPH COMPRESSOR, TRIUMPH ICE MACHINE CO., CINCINNATI, OHIO, U. S. A.

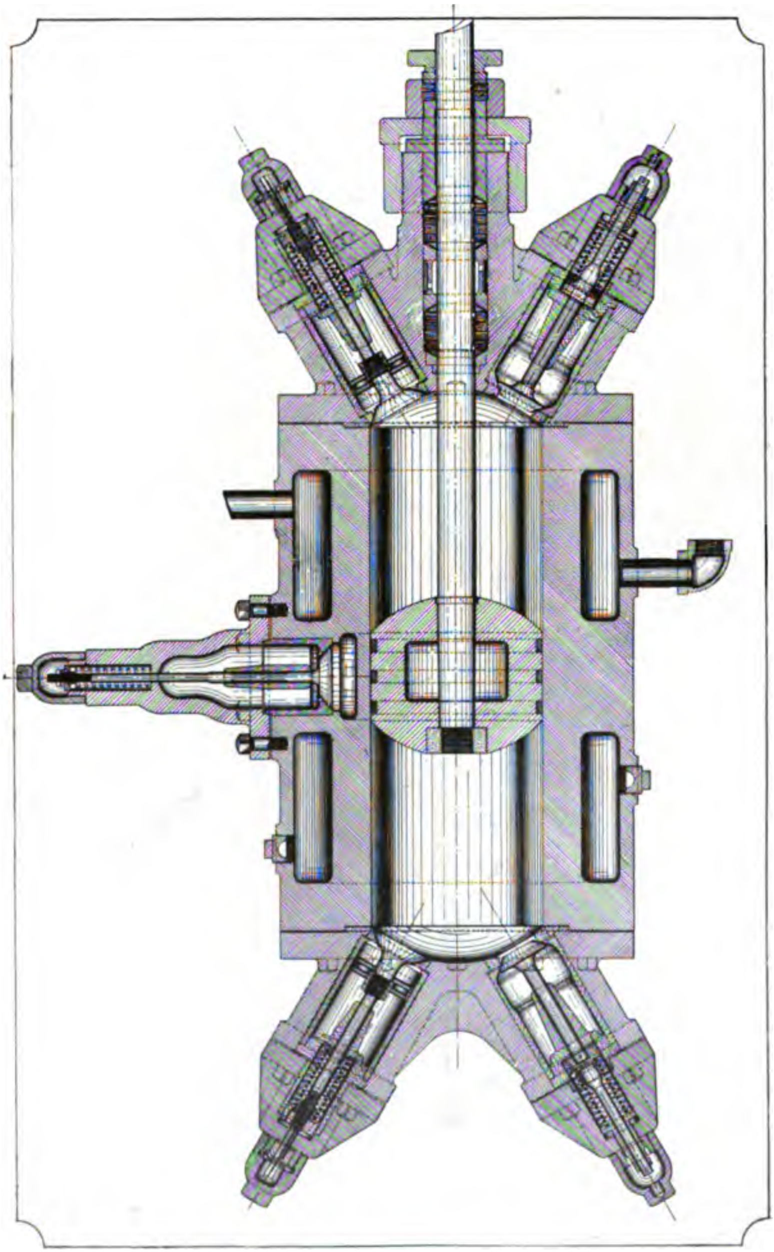


FIG. 182.—SECTION OF TRIUMPH ICE MACHINE CO.'S COMPRESSOR CYLINDER, CINCINNATI, OHIO, U. S. A.

An inspection of the great works of the Fresh Food and Ice Co., of Sydney, N. S. W., affords a splendid example of the progress that has been made during the past few years in improving and perfecting refrigerating machinery in its various applications to the needs of commerce. The great

FIG. 183.—HERCULES MACHINE IN NEW SOUTH WALES FRESH FOOD AND ICE CO.'S WORKS, SYDNEY, N. S. W.

work of the late Mr. T. S. Mort (referred to in the historical chapter), is still in evidence, and is continually expanding. This company, of which two of Mr. Mort's sons are directors, is the largest company of the kind in the southern hemisphere, and in its way unique. It runs refreshment rooms, and it has a large railway and shipping business, both

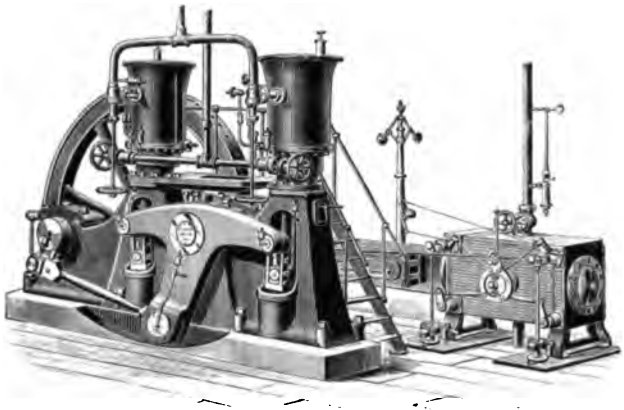


FIG. 184.—LATEST DESIGN HERCULES LARGE COMPRESSOR.

FIG. 185.—LATEST DESIGN HERCULES MACHINE, STEAMSHIP PATTERN.

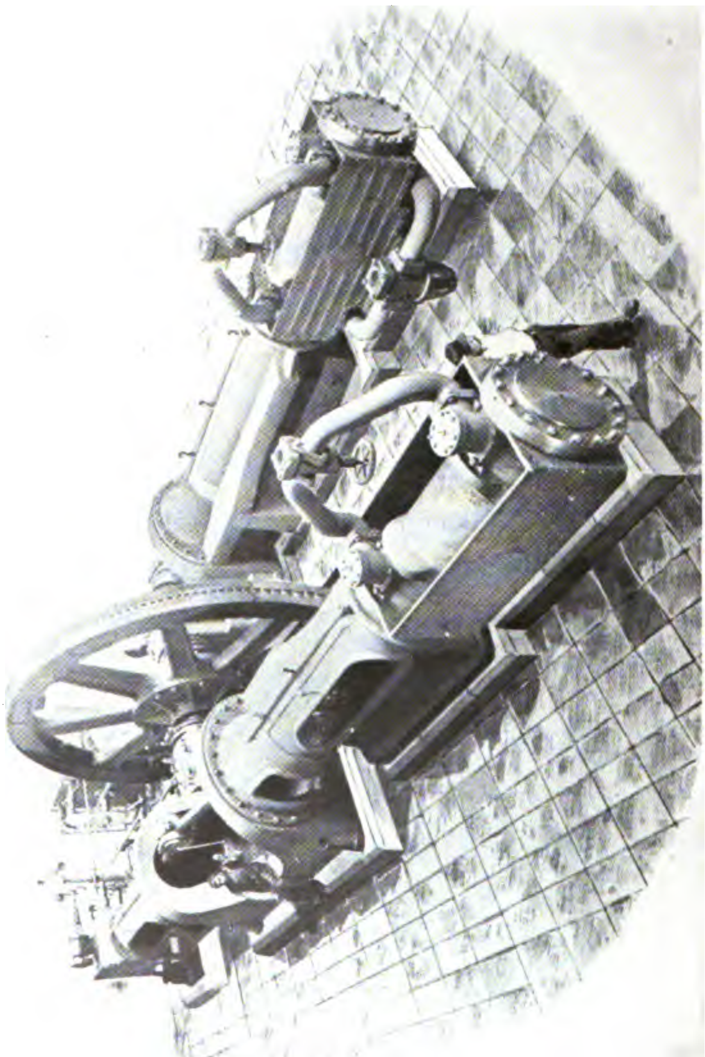


FIG. 186.—THE BALL COMPRESSION MACHINE, ICE AND COLD MACHINE CO., ST. LOUIS, MO., U. S. A.

domestic and foreign, in ice, fish, poultry, rabbits and hares. The New South Wales railways run into its premises, carrying its own refrigerating vans. Its milk tanks, at headquarters alone, have a capacity of over 30,000 gallons, and in one month it has shipped 80,000 frozen sheep to London.

The freezing machinery of this company's principal works includes two compound compressing plants made under the Lock patents, one De La Vergne machine and one Auldjo machine, all four machines being built by the Morts Dock Co., of Sydney. There is also one De La Vergne machine, made in New York, besides the latest addition to the plant, which is a 70-ton Hercules machine, with compound tandem engine. Fig. 183 is a view from one angle of the principal engine room. The Hercules machine is the prominent feature, while the De La Vergne machines will be noticed in the rear. Since the World's Fair at Chicago, Australia and New Zealand have been so well exploited in the interests of the Hercules machine, that it is now running in much greater numbers than other makes of refrigerating plants throughout the colonies, and as Figs. 72 and 73 are only diagrams, more justice is done to its importance by this later illustration and Figs. 184 and 185, the former showing the latest design of large machines, the latter the steamship pattern, as made by C. A. MacDonald, Chicago and Sydney.

The illustration No. 186 deserves notice for several reasons. It is a perspective of a gigantic machine, as shown by the comparative size of the men alongside, and is rated as equal to 725 tons refrigeration. The builders are the Ice and Cold Machine Co., of St. Louis, Mo., and their design presents differences in detail from any machine so far referred to. It is a straight line machine, but it embodies the arrangement advocated on pages 130 and 131—with a right-angled connection—in having a straight shaft and only two bearings; the fly-wheel being in the center, and a crank at each end. Where it differs from the average straight line machine is in the adoption of cross-heads, guides and connecting rods, to both the steam and ammonia ends; two connecting rods, side by side, being connected to the same crank pin. This, of course, adds to the length of the machine, and increases the frictional losses; but there are, no doubt, good and

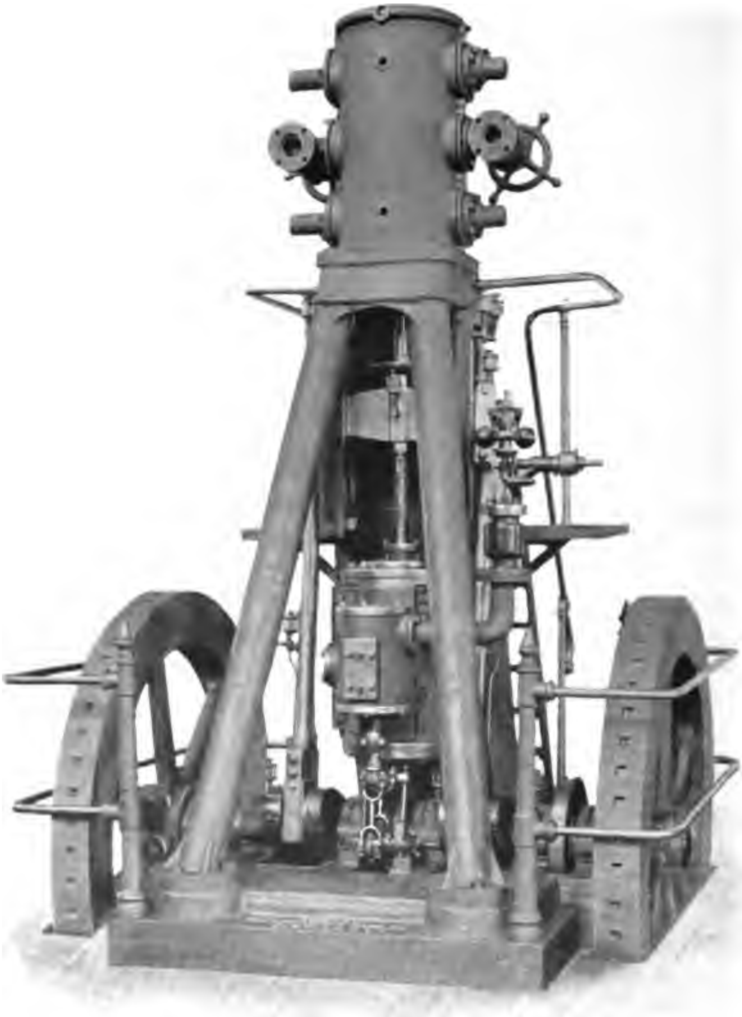


FIG. 187.—BUFFALO REFRIGERATING MACHINE CO.'S COMPRESSOR, BUFFALO, N. Y., U. S. A.

weighty reasons for adopting this arrangement, instead of the more usual one of connecting up the steam and compressor pistons to one cross-head only, and with one connecting rod to each side. One of these reasons is obvious, and that is, it enables either of the ammonia cylinders to be discon-

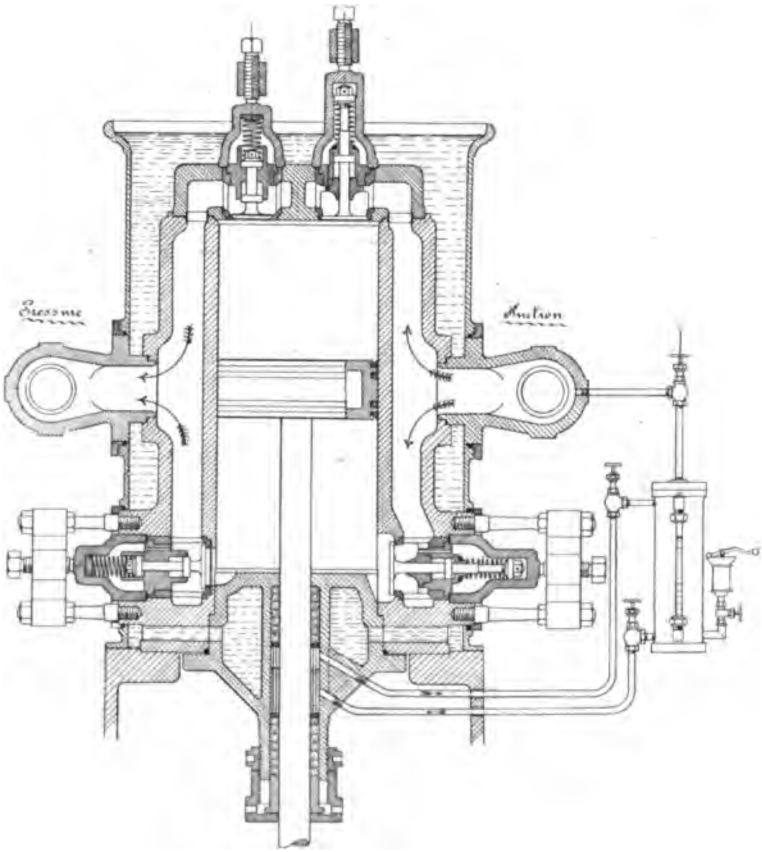


FIG. 188.—SECTION BUFFALO REFRIGERATING MACHINE CO.'S COMPRESSOR CYLINDER, BUFFALO, N. Y., U. S. A.

nected without requiring the engine on the same side to be stopped also.

A further reference to the cut will show that the valve is located on the cylinder, being a gravity valve without springs. It works over a plunger, the cushion of gas for closing same being regulated by a needle valve, which regu-

lates the compression or vacuum in the chamber formed by the valve and plunger, the suction valve being directly opposite to the discharge valve (see Fig. 165).

Fig. 187 is a perspective view of the latest design 25-ton vertical straight line machine, manufactured by the Buffalo Refrigerating Machine Co., Buffalo, N. Y. It is double-acting. The ammonia compressor and steam cylinder are in alignment and bolted to a rigid cast iron frame, mounted on a heavy and substantial bed plate, in one piece. The machine is therefore self-contained. The form of construction, as illustrated by this machine, has been exhaustively treated in Chapter XV of this work.

This compressor may be operated by an engine of either the slide valve, automatic cut-off or Corliss pattern. The clearance in cylinder is reduced to a minimum.

The piston is provided with patented self-adjusting packing rings, one at the top and one at the bottom end. The pressure of the ammonia gas acting upon the conical surface of the ring expands the same in all directions outward against the wall of the cylinder, forming a perfectly tight joint.

The pressure and suction valves are of ample area to handle the gas without wire-drawing, and their construction is such that they leave but little or no useless space inside of the cylinder, in which the compressed gas can collect. The valves are made of forged steel, case-hardened on seats, and are ground to a perfect seat. They have long guiding surfaces and are arranged with cushioning chambers to relieve them from undue strain, prevent slamming and bring them gently and noiselessly to their seat. The stem of suction valve is provided at the bottom with a collar, which prevents the valve from dropping into the cylinder in case the nut on top of the valve stem should get loose. The cages in which the valves work are made of cast steel, and are so arranged, as will be seen in the section Fig. 189, that they can quickly and easily be removed and replaced without disturbing any other connection.

The stuffing box is long and is so arranged that between the upper and lower packing an oil chamber is provided, which is automatically supplied with oil from the oil tank, as shown

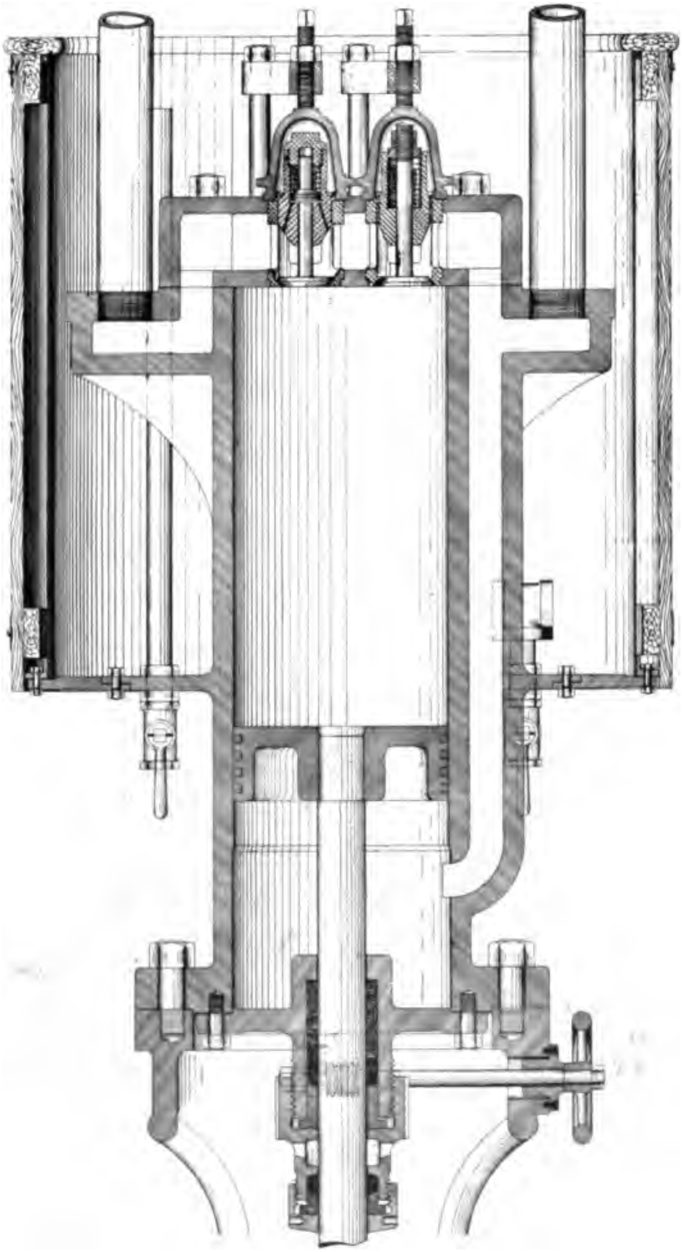


FIG. 189.—BOYLE COMPRESSOR, PENNSYLVANIA IRON WORKS CO., PHILADELPHIA, U. S. A.

in cut. The operation of the oil arrangement is as follows: The oil tank is supplied as often as necessary with oil by the hand pump attached to the tank. The lower end of the tank is connected to the lower part of oil chamber in the stuffing

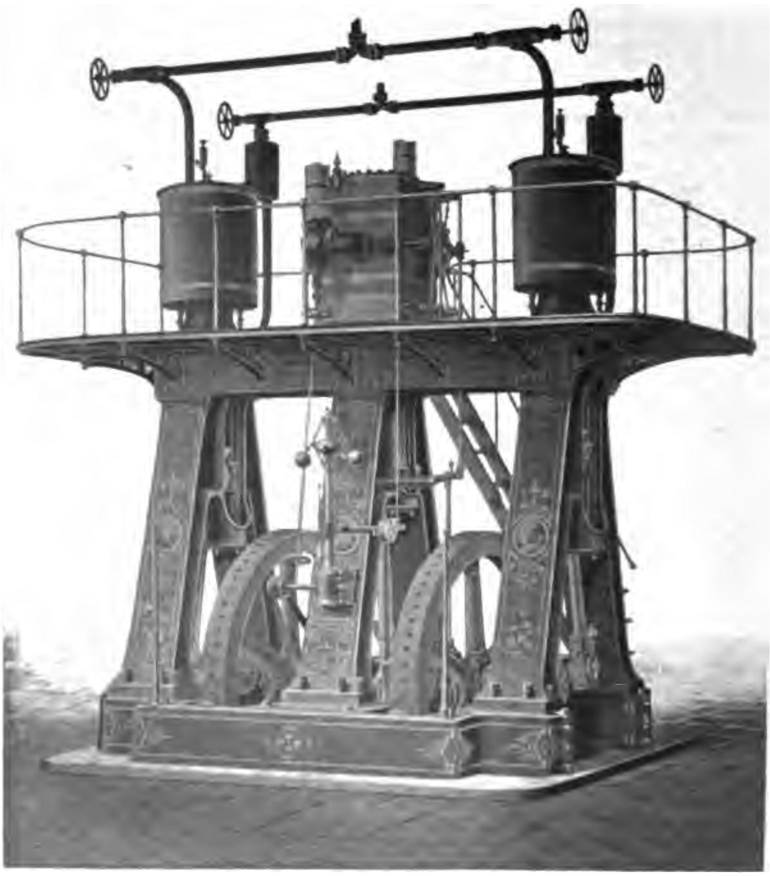


FIG. 190.—BOYLE SINGLE-ACTING MACHINE WITH VERTICAL ENGINE, PENNSYLVANIA IRON WORKS CO., PHILADELPHIA, U. S. A.

box, and the upper end of chamber is connected to the upper end of oil tank; a connection also is made from upper end of oil tank to suction valve of machine. By this means the oil in oil tank will be under suction pressure of the ammonia gas

on the top and bottom side, and as the oil tank is placed above the oil chamber in stuffing box, the oil will flow into the latter by its own gravity, and any leakage of ammonia from the ammonia cylinder through the first layers of packing into the

FIG. 191.—BOYLE SINGLE-ACTING MACHINE WITH HORIZONTAL ENGINE, PENNSYLVANIA IRON WORKS CO., PHILADELPHIA, U. S. A.

oil chamber of the stuffing box will be drawn into the suction pipe of the machine, and consequently the pressure in stuffing box, it is claimed, can never exceed the suction pressure under which the machine is working. The quantity of the

oil fed to stuffing box is regulated by the valves on pipes communicating with the oil tank. The oil, adhering to the piston rod, finds its way into the cylinder in sufficient quantity to lubricate same.

The gas cylinder and its top and bottom head is surrounded by a water jacket for removing the heat of com-

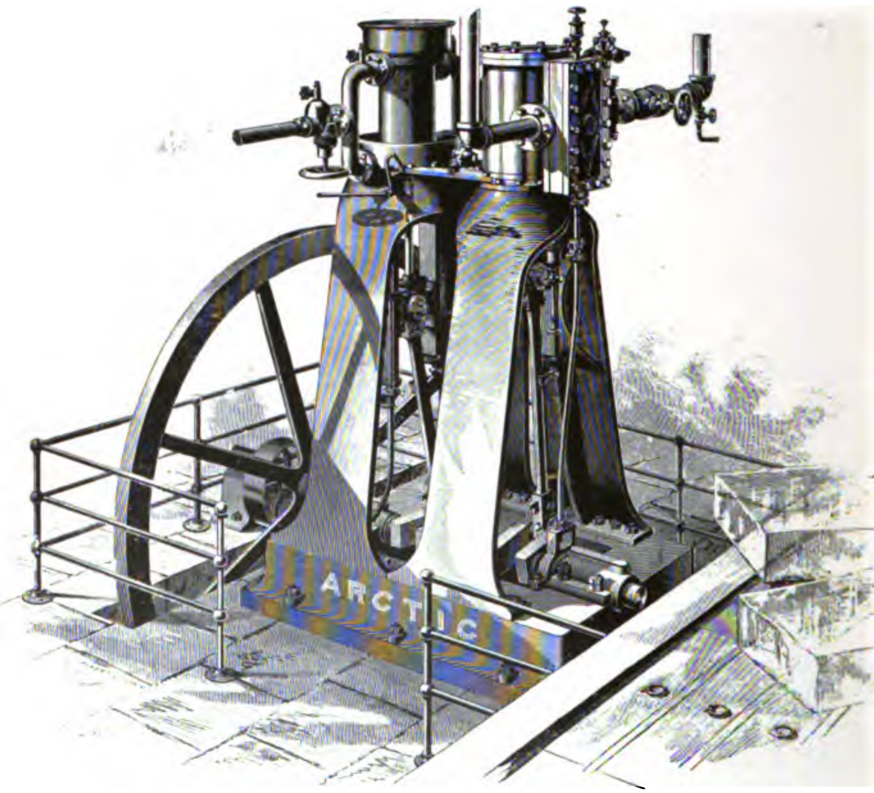


FIG. 192.—TYPE OF ARCTIC AMERICAN MACHINE BUILT IN 1879.

pression as far as possible. The clearance space between the cylinder heads and piston is reduced to a minimum, only the thickness of the sheet packing being allowed.

Reference is made on page 132 to the Boyle vertical machine, as built some twenty years ago, in comparison with same machine as built by the Pennsylvania Iron Works Co., Philadelphia, U. S. A., and illustrated on page 133.

The distinctive features of this modern type of the Boyle machine lie in two vertical single-acting compressors in combination with either a vertical or horizontal engine. See Fig. 189, being a section of the cylinder, and Figs. 190 and 191, showing single-acting compressors, with vertical and horizontal engines, respectively.

The compressor valves are inclosed in removable cages, both suction and discharge of which are located in the upper

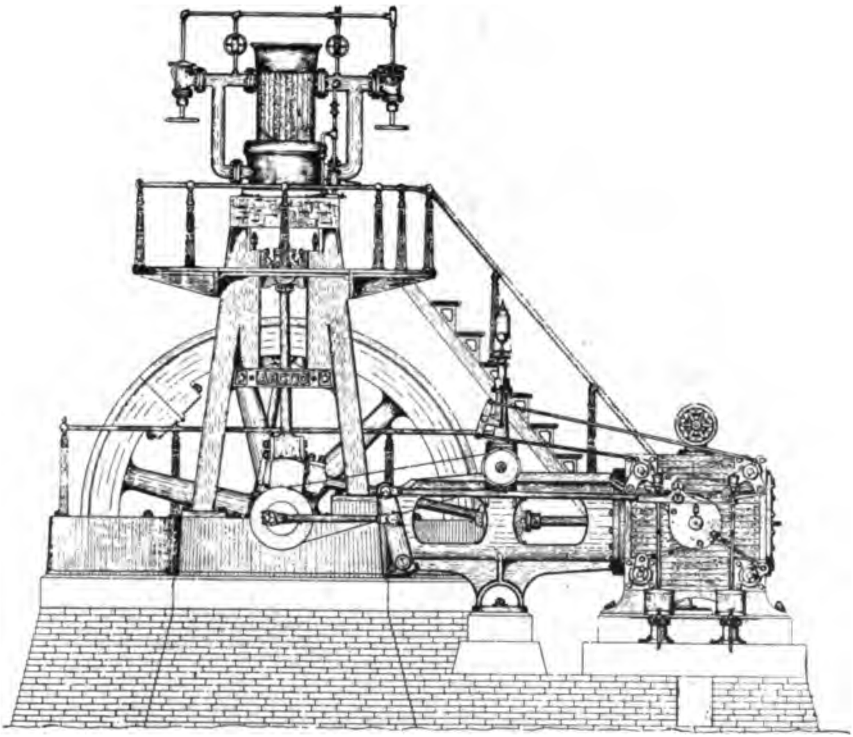


FIG. 193.—SECTION ARCTIC AMERICAN MACHINE AS BUILT IN 1900.

head, being held in position by cross-bars and a single set screw on top of each, the head having a division in the center, the gas entering through its pipe, which is screwed into a pocket extending from the body of the cylinder, and communicating with the inlet chamber, and through its valve entering the cylinder during the downward stroke of the piston. Upon the return stroke the gas is compressed until it equals

the pressure within the condenser, the discharge valve in the opposite side of the head then lifting and allowing the discharge of the gas through the valve and communicating chamber to the discharge pipe from the compressor. The suction chamber also communicates with the lower end of the compressor cylinder, filling the same with gas during the upward stroke of the piston, and allowing its exit during the downward stroke thereof. The upper portion of the cylinder is surrounded by water jacket, having inlet and outlet openings provided for the flow of water from the same, taking up a portion of the heat due to the compression of the gas, and keeping the compression valves and different parts at a temperature to not interfere with their proper operation.

The compressor piston is of the solid type, having a number of snap rings, the tension of which makes them tight enough to prevent the leakage of gas past them.

The stuffing box has an evaporating pressure only upon it, is easily kept tight, and consequently there is little wear or tear on the rod.

Owing to the single-acting feature, the clearance can be reduced to a minimum, the compressor piston traveling as close to the head as possible without touching same. The piston and valves being perfectly balanced also, exert no side wear upon the cylinder or other parts, and present the most desirable features for continued service.

The Arctic machine, built at Cleveland, Ohio, U. S. A., has been on the market since 1879, and machines of that date are still in use. It was one of the first of this class of machines to come into general use in America. Fig. 192 is a perspective of an Arctic machine, as built in 1879; Fig. 193 is a section of machine, as built in 1900, by the Arctic Machine Co.

The machine is of the double-acting ammonia compressor type, built either with a vertical steam engine and vertical compressor or with a horizontal steam cylinder and two vertical compressors.

The construction of this machine has changed in many ways, *i. e.*, the large fly-wheel has given place to one more in proportion, and usually placed between the columns; when placed on the outside the shaft has an outside bearing. The compressor valves are now fitted in cages; formerly the head

of compressors had to be removed to get at them. The stuffing box of compressor is deeper and fitted with oil sleeves. Corliss valve motion has taken place of the slide valve, while connecting rods, cross-heads, piston, etc., have been made to conform to modern practice.

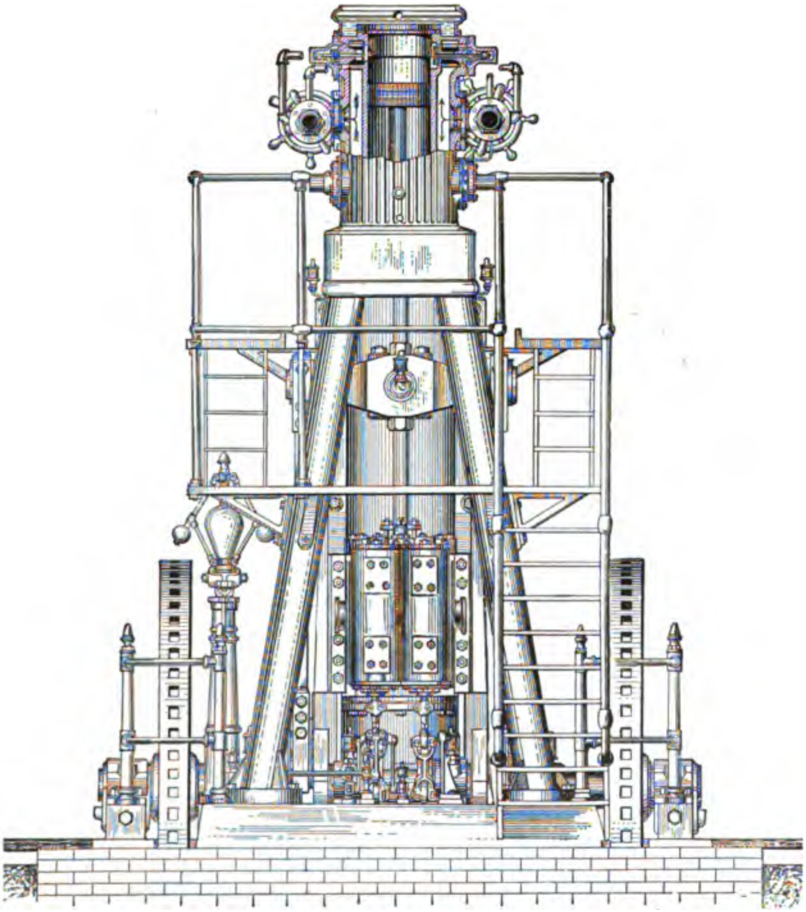


FIG. 194.—SECTION OF CASE COMPRESSOR, CASE REFRIGERATING MACHINE CO., BUFFALO, N. Y.

Fig. 194 is an illustration of a refrigerating machine built by the Case Refrigerating Machine Co., of Buffalo, N. Y. These machines are of heavy build, and occupy comparatively small floor space. The peculiarity of the construction of the

machine is that both the steam and compression cylinder piston rods are connected to the same cross-head, which works between the two cylinders. The steam cylinder is below and in a direct line with the compression cylinder. This allows a direct push and pull on the piston rods, calculated to remove all strain from the crank shaft and connecting rods, and thus reduce the friction to a minimum.

A water jacket surrounds the compression cylinders, where a small stream of water is kept running to cool the compressor when in motion.

FIG. 195.—THE BARBER TYPE AMERICAN COMPRESSION MACHINE.

The compressor suction and discharge valves work horizontally, which allows a very small pocket for compressed gas, and reduces the clearance to a minimum.

The machine illustrated by Fig. 195 is manufactured by the A. H. Barber Manufacturing Co., Chicago, U. S. A. It is of the horizontal type, with a double-acting compressor. It is built with a box frame, with a center crank for those run by belt, and a tangye frame or side crank when directly connected to Corliss or slide valve engine. The shaft, pulley and fly-wheels are all in proportion to the size of the compressor. The cylinders are let down into the frame. A flat locomotive guide is used, thereby giving the machine a deep and rigid frame, so that it is impossible for the cylinder to get out of line.

The cylinder and valves are entirely surrounded by water.

The valves and seats are made of tool steel, and both are hardened so as to prevent pitting and to increase their wearing efficiency to the utmost. The valves are easily removed for inspection, without breaking or disturbing any other joint. The piston is made as light as possible, and provided with metallic packing rings. The stuffing box is perfectly sealed, having a double packing, with an oil chamber in the center. The lubricator is so arranged that it oils the cylinder, valves and piston rod.

In the suction conduit, close to the compressor, is placed a strainer, which prevents scales from the system getting into the compressor. The clearance is reduced to a minimum, and the connecting rod is so arranged that any wearing on the crank shaft or guide can be easily adjusted.

The Challoner machine, illustrated by Figs. 196, 197 and 198, belongs to the inclosed type described in a former chapter.

The frame is cast in one piece, having two heavy ribbed flanges, and secured to a bed plate cast in one piece, strongly arched where frame rests upon it. Each end of the frame is provided with heavy circular removable flanges containing long babbitted bearings for the crank shaft to rest in, and extra long stuffing boxes with glands and nuts to prevent any leakage of gas or oil around shaft. Within the frame the bearings for the crank shaft are bolted in place so as to be readily removable. The case is supplied with a charge of oil, so that all working parts run in same, to insure lubrication without exterior oilers or lubricators. The top of the case is faced off true and bored out to receive the compressor cylinders, which are sleeve castings and can be removed and replaced in case of necessity without renewing the case or frame.

The crank shaft for the small machines is a solid steel casting, while for the larger sized machines it is a solid steel casting for each outboard end, with center crank of cast steel, all put together with turned faced male and female joints, securely bolted with turned bolts in reamed holes. The connected parts of shaft form a very large bearing surface in the journals inside the case, to insure permanent

alignment of the shaft and minimum wear of the journals. The larger sizes of the machines are provided with two and the smaller sizes with one extra heavy large band fly-wheels.

The connecting rods are made of hammered iron, the upper ends being bored out and provided with hardened steel sleeves for bearings on the piston pins. The lower

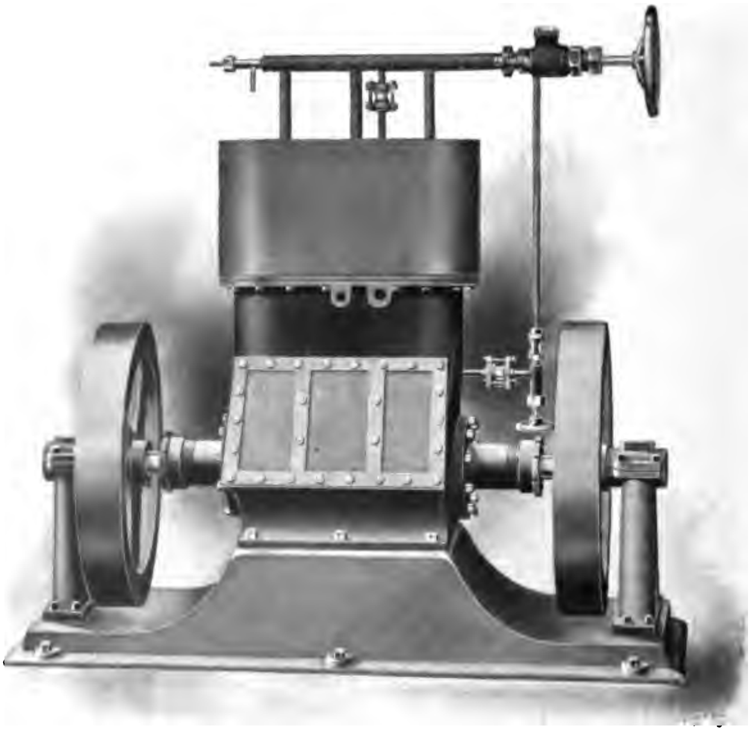


FIG. 196.—THE CHALLONER TRIPLE CYLINDER, SINGLE-ACTING COMPRESSOR, GEO. CHALLONER'S SONS CO., OSHKOSH, WIS., U. S. A.

ends of the rods are threaded and passed through the yokes of the crank stubs, having keys in the stub yokes to prevent turning of the rod out of line with the crank and piston pins.

The larger size of machines are provided with safety heads, the action of which, should the pistons be set too close to the heads, would be to lift and prevent possibility of knocking out a head.

The suction valves are placed in the pistons, and the discharge valves are placed in the safety heads or in the false heads. Both the suction and discharge valves may be removed by taking off the pump heads and without disconnecting pipe connections.

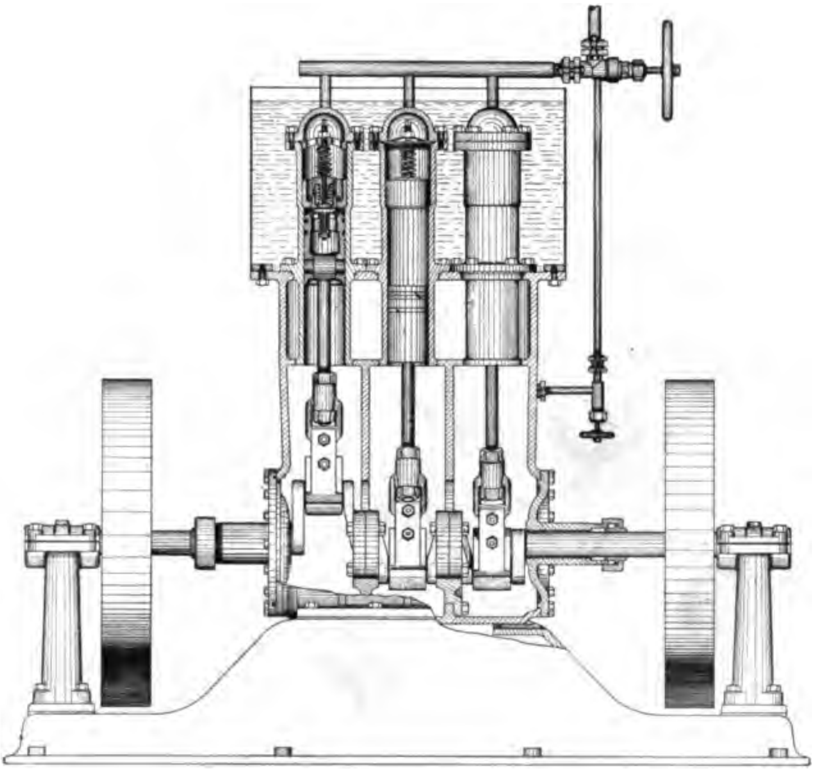


FIG. 197.—THE CHALLONER TRIPLE CYLINDER, SINGLE-ACTING COMPRESSOR SECTION.

The suction connection is made to the case below the cylinders so that the case is kept cool by the return of the low temperature gas. The discharge connections are made to the cylinders above the safety heads. Both connections are provided with suitable stop valves, and by-pass connection is arranged so that the machine can readily be reversed to pump the gas from the high pressure to the low pressure

side of system. A purge valve is also placed on the discharge connection so that the case may be pumped out for opening and examining, merely with a few turns of the crank shaft, and all air entering the case when opened can be discharged to the atmosphere, thereby preventing accumulation of permanent gases in the system.

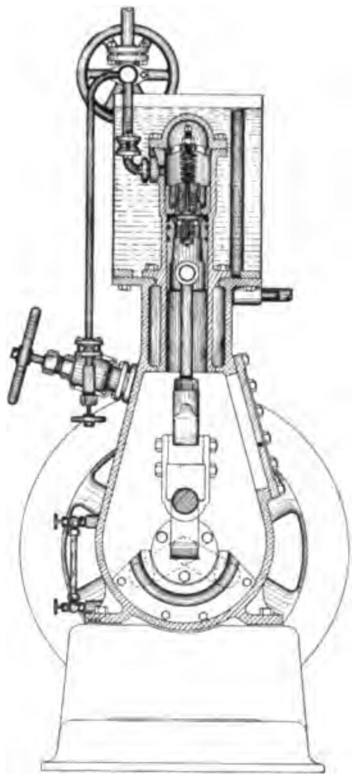


FIG. 198.—THE CHALLONER TRIPLE CYLINDER, SINGLE-ACTING COMPRESSOR, END VIEW.

On page 124 reference is made to the many devices that have been brought forward by the inventive skill of engineers, for the purpose of distributing the work of a compressor piston more evenly, in relation to the motive power. In Fig. 199 there is an elevation of a quite recent design, known as the "Ideal" machine, and built by the Ideal Refrigerating and Manufacturing Co., of Chicago. It will

be seen that the diameter of the crank pin circle is much greater than the stroke of the piston; and that, as the con-

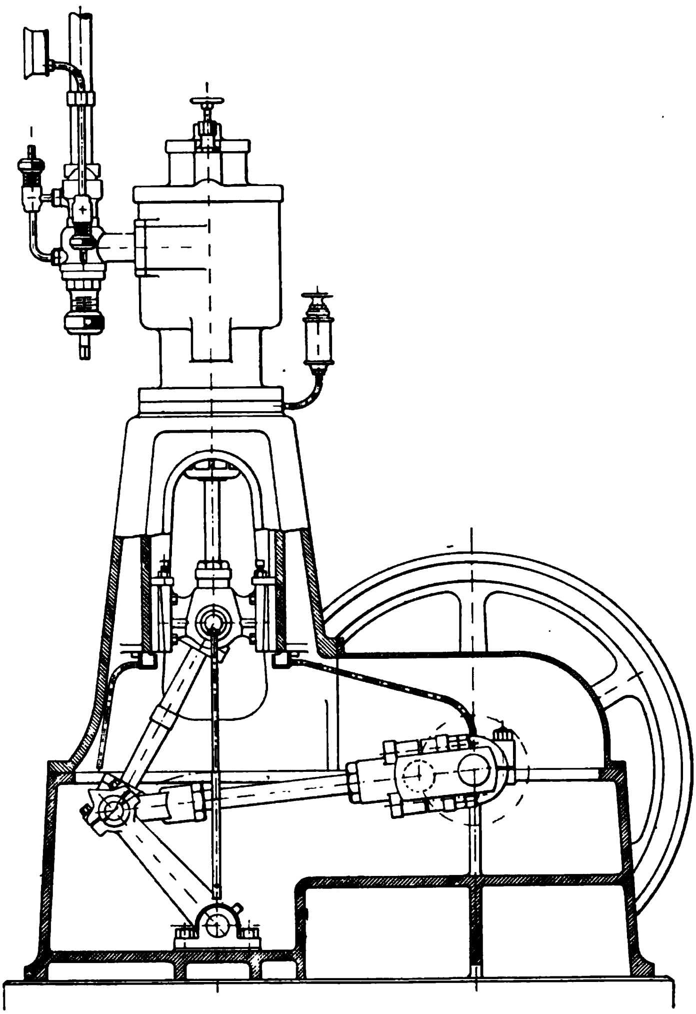


FIG. 199.—SECTION OF IDEAL REFRIGERATING AND MANUFACTURING CO.'S MACHINE, CHICAGO, U. S. A.

necting rod pulls the two members of the toggle joint toward the vertical position (where the joint pins are in a straight line) the force available to move the piston gradually

approaches the infinite, apart from the toggle action of the crank itself. Fig. 200 shows graphically how the moments of resistance to the turning of the crank shaft differ from an ordinary direct connection; the cam-shaped diagram drawn in full line representing, in the radial lengths from the crank pin circle, the resultants from the theoretical compressor diagram above. The dotted figure corresponds with that resulting from a direct connection from the crank to the piston rod cross-head, like Figs. 98 and 99.

As an effect of this toggle action it is apparent that the first half of the compressing stroke, from position 0 to position 3, is made by the piston, while the crank pin travels

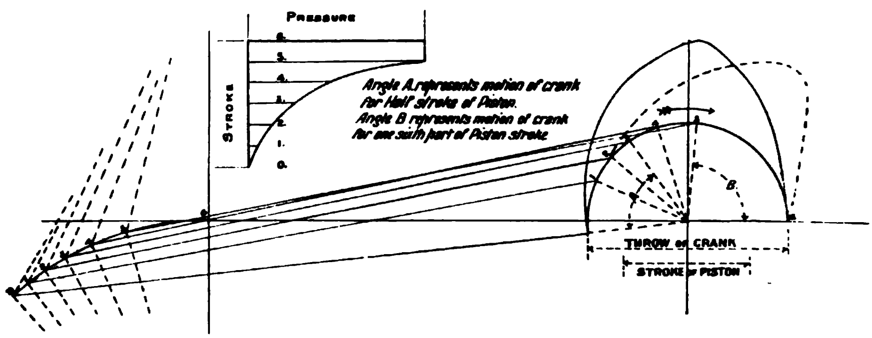


FIG. 200.—DIAGRAM SHOWING EFFECT OF THE MOTION IN "IDEAL" MACHINE.

through less than one-sixth of a revolution; but that during the second half of the piston's stroke, from position 3 to 6 (where most of the work is concentrated), the crank pin moves through rather more than two-sixths of its course.

The manufacturers of this machine claim to know from actual experience that the effect the intermitting motion of the cam-head on piston has on the valve is to prolong the life of same more than double, compared to that in an ordinary crank motion machine. This is due, it is argued, to the prolonged stop caused by the crank passing over the dead center, and the toggle being in a straight line with the piston rod at the same time. The advantage claimed is that it gives the valve ample time to get seated, and all the gas is discharged

from the cylinder, and not drawn back into it on the return stroke, thereby developing a very high efficiency.

Owing to the throw of the crank being greater than the stroke of the piston, the stress, and, therefore, the friction

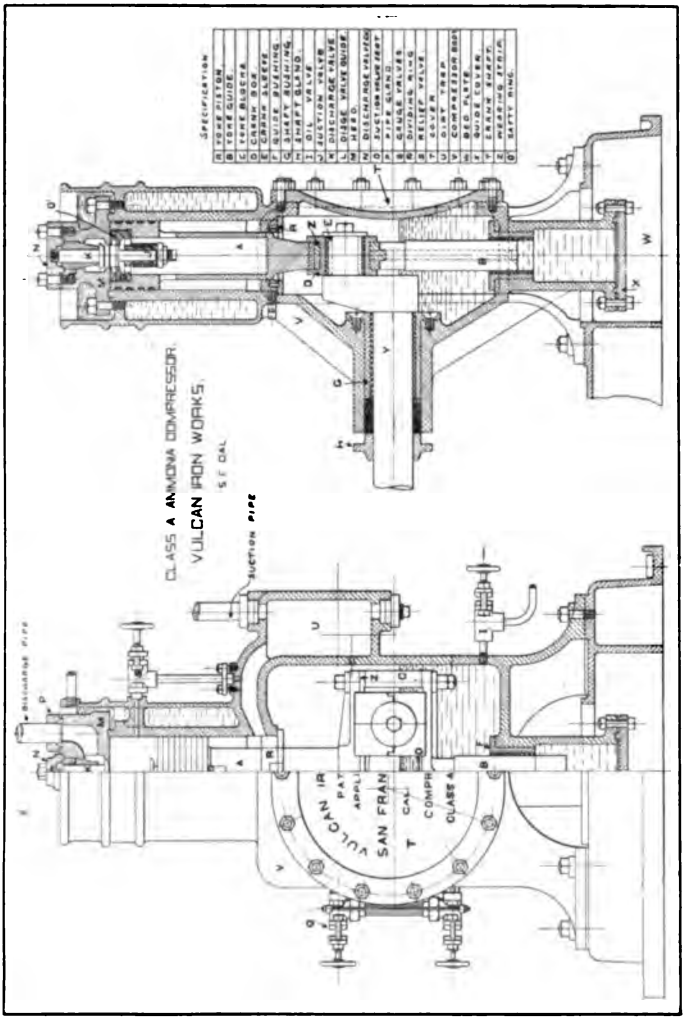


FIG. 201.—SECTIONAL VIEW "VULCAN" COMPRESSOR, CLASS "A."

and wear on the crank pin, is reduced to that extent by the toggle device.

Figs. 201, 202 and 203 illustrate the class "A" and class "B" "Vulcan" refrigerating and ice making machines, man-

ufactured by the Vulcan Iron Works, San Francisco, Cal. These machines are furnished in sizes up to ten tons re-

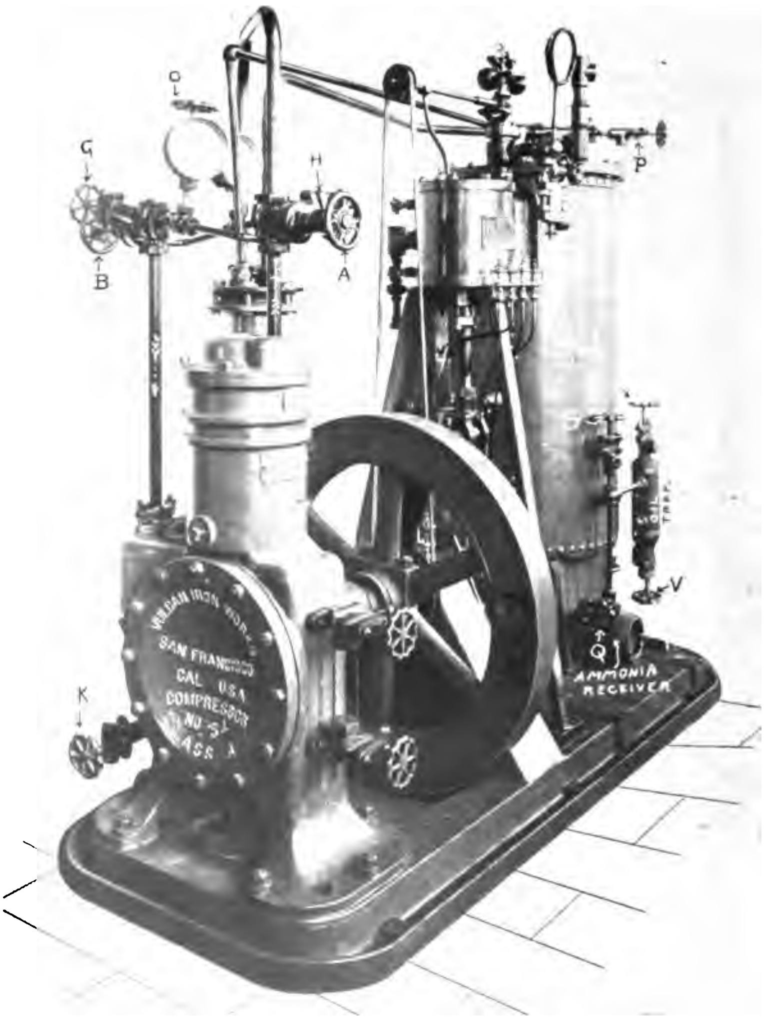


FIG. 202.—"VULCAN" COMPRESSOR, CLASS "A."

frigerating capacity. Machines of larger sizes are of the horizontal double-acting type (class "D").

These special styles were designed to meet the demands for small machines that would embody simplicity of design, construction and operation, and include a number of distinctive features.

The compressor is vertical, single-acting, and of the inclosed type, the working parts being automatically lubricated

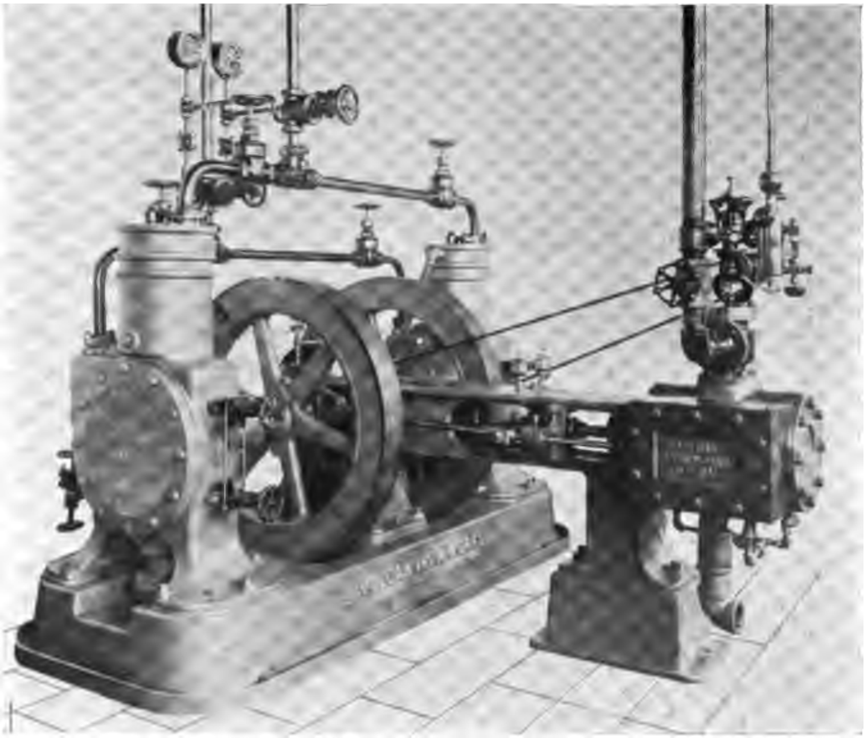


FIG. 203.—"VULCAN" COMPRESSOR, CLASS "B," WITH STEAM ENGINE.

by the oil in body of machine. (See sectional view, Fig. 201.) The cylinder opens into crank chamber, the sides of which form the supporting frame, thereby bringing the cylinder and shaft close together, and doing away with the long connections otherwise made necessary by piston rods and crossheads, and making this compressor a compact, strong and

accessible machine. The crank is forged on end of heavy steel shaft, which passes through stuffing box in side of crank chamber. The crank pin is of special construction, having hardened steel sleeve held in place by collar. The piston is operated by a crank working in box that slides in a yoke that is made part of the piston, the yoke having a guide at bottom, and being guided by the piston at the top. The crank chamber is provided with a removable cover. When machine is in operation the body of machine is filled with oil to a point just above stuffing box of crank shaft, the height of the oil being shown by a gauge glass, the oil acting as a lubricant for the moving parts and also as a seal for stuffing box of crank shaft. The body of machine is separated from the ammonia cylinder by a dividing or packing ring (R), through which the trunk or piston works, in such manner as to admit only sufficient oil to lubricate cylinder. The suction valve, which is fitted with a safety cage, is placed in center of piston (the ammonia gas enters body of machine below piston) and the discharge valve is placed in cylinder head.

A dirt trap (U) is attached to each compressor body, into which the ammonia suction pipe discharges, intended to prevent the passage into ammonia cylinder, of any scale, dirt, etc. The relief valve (S) is for convenience in starting the machine. Wearing parts are supplied with removable bushings.

A class A machine has only one ammonia cylinder. (See Fig. 201.) A class B machine has duplex ammonia cylinders of the class A type.

The class A machines are self-contained, *i. e.*, the ammonia compressor, ammonia condenser, steam engine, ammonia and oil receivers, ammonia gauges and pipe connections (also brine pump, if required,) are all placed on one bed plate, thus making a very compact arrangement, and especially suitable for use on steamships.

Figs. 204 and 205 show in perspective and cross-section the construction of the Stallman compressor, manufactured by the Creamery Package Manufacturing Co., of Chicago, Ill. This machine is of the vertical single-acting, water-jacketed type, and made in sizes from two to ten tons refrigerating capacity. The lower parts of the cylinders are cored out, so as

to form a series of ports leading from the suction inlet around the piston and into the cylinders, when the pistons are at the bottom or limit of their downward stroke. The filling of the cylinders having been partially effected by the passing of the gas through the suction valves in the pistons during their downward stroke, is thus at the very end of the stroke fully

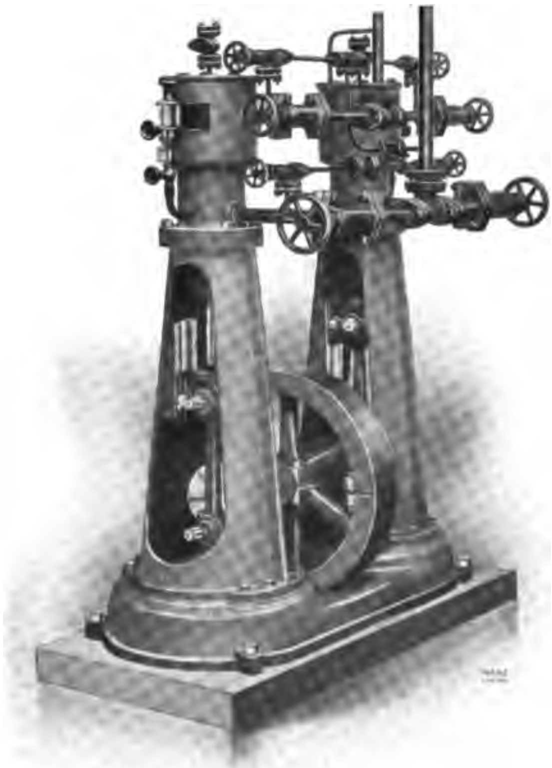


FIG. 204.—PERSPECTIVE VIEW STALLMAN COMPRESSOR, CREAMERY PACKAGE MFG. CO., CHICAGO, U. S. A.

completed, and the full evaporating pressure secured in the cylinders by the unobstructed passing of the gas through these ports. The upper part of the cylinder is enlarged, and upon the shoulder thus formed rests the discharge valve seat, which is made of tool steel and is pressed into position before the finishing cut is taken. It is then bored out with

the cylinder and forms a part of the cylinder wall. Immediately above the valve seat, connected with the enlarged part of the cylinder and branching off at right angles, is the outlet port, which receives the discharge pipe.

The discharge valve is made of steel. It is turned up from the solid, with a disc-like bottom larger than the bore of the cylinder, thus extending over and resting upon the tool steel seat above described.

On its upward stroke the piston passes through the discharge valve seat and comes into metallic contact with the valve itself, discharging completely the contents of the cylinder past the valve, and leaving no gas to re-expand. There is therefore absolutely no clearance and consequently no loss of efficiency from this source.

The valves, being large, have but slight movements and practically instantaneous action, and at the same time give very large areas of openings that permit the rapid passing of large volumes of gas. The valve and cylinder construction of this compressor should give the maximum results for power expended.

Attached to and forming part of the discharge valve is a band-like extension that takes the place of a valve stem, the enlarged portion of the cylinder forming the guide for the valve. In the center of the discharge valve is a boss or center, around which is placed a spiral spring. This spring is provided with a screw, passing through the cylinder head, for adjusting its tension, not shown by cut. The piston is fitted with cast iron snap rings, turned to bore of cylinders.

In the shell of the piston is the suction valve guide, held in position by the steel valve seat, which is threaded to and surrounds the upper part of the piston shell.

The cylinders are mounted upon frames containing the shaft bearings and guides to bring all strains directly upon the frames and not upon bearings in a separate bed plate; in this construction the rigidity of the alignment is assured. A heavy box pattern bed plate securely ties the frames in position, making a compact and yet convenient arrangement throughout.

The construction permits of operating the compressors independent of each other where conditions of varying tem-

peratures and consequent varying back pressures prevail, such as in plants for both ice making and refrigerating, and

FIG. 205.—CROSS-SECTION STALLMAN COMPRESSOR, CREAMERY PACKAGE MFG. CO., CHICAGO, U. S. A.

where freezing rooms are used in connection with ordinary cold storage. Independent suction connections can be made to the compressors under such circumstances.

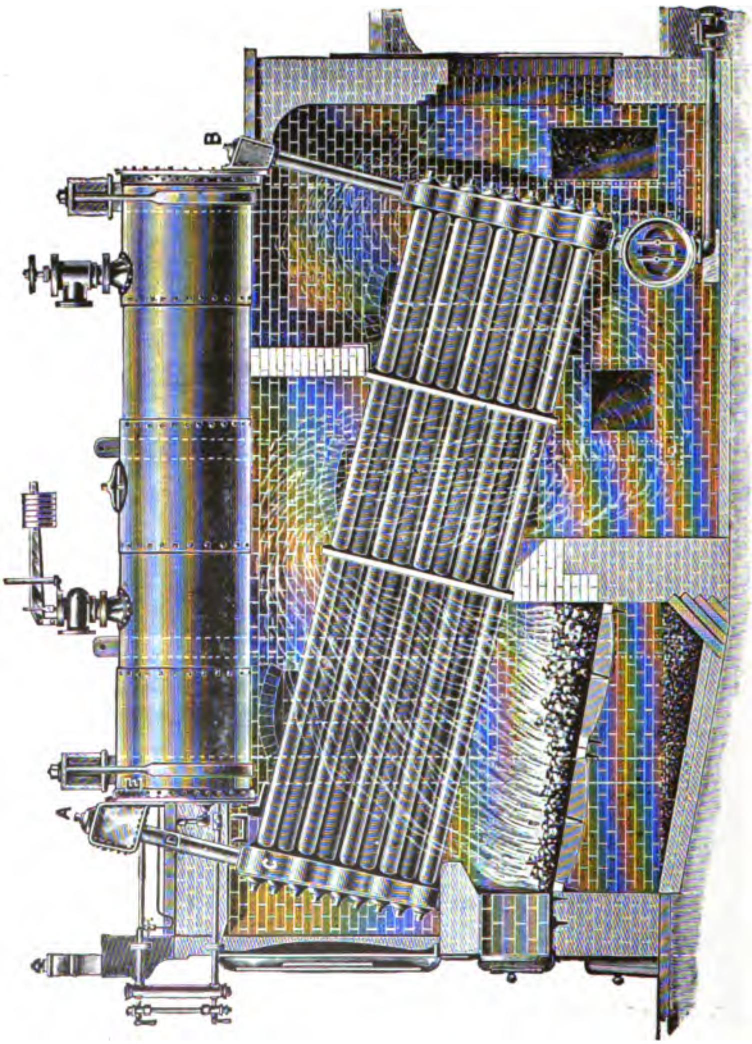


FIG. 206.—IMPROVED "PREMIER" WATER TUBE STEAM BOILER.

FURTHER REMARKS CONCERNING WATER TUBE BOILERS.

It would be difficult to decide whether the water tube boiler is at present creating a greater revolution at sea or on shore; possibly it is more so in connection with steam vessels. One of the many lines of mail steamers trading to Sydney (now said to be the fourth most important port for shipping in the world) has been carrying the Belleville boilers for years, and they have obtained a footing in the English, American and foreign navies.

For present purposes we are more concerned with land types, and as no illustrations appear in the sub-section com-

FIG. 207.—FIRE TUBE AS AFFECTED BY SOOT AND DIRT.

FIG. 208.—WATER TUBE AS AFFECTED BY SOOT AND DIRT.

mencing on page 205, Fig. 206 will give a good idea of how nearly every inch of such boilers is utilized for heating surface. In this figure the removable covers for scaling the tubes are clearly seen, as well as the doors in the brick work to enable the sooty deposit to be removed from their external surfaces.

As Figs. 122 and 123 illustrate the advantages of fire tubes in connection with the deposit of scale, it is only fair to give an illustration by which the advocates of water tube boilers show their great advantages with regard to the deposit of soot and dirt from the fire.

It is maintained, and is no doubt true, that if the draft is weak and the ordinary tubes are not attended to, they

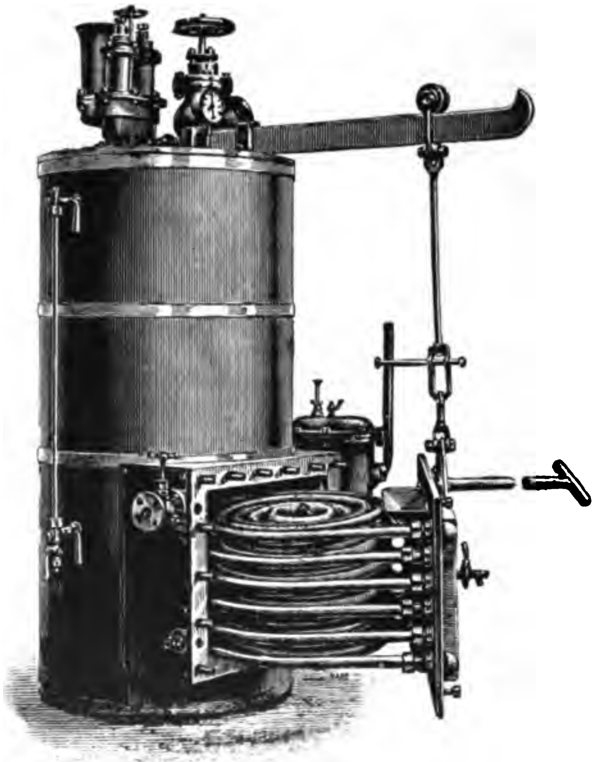
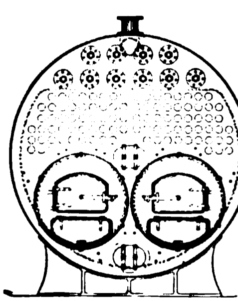


FIG. 209.—EVAPORATOR FOR WATER HEAVILY CHARGED WITH MINERALS.



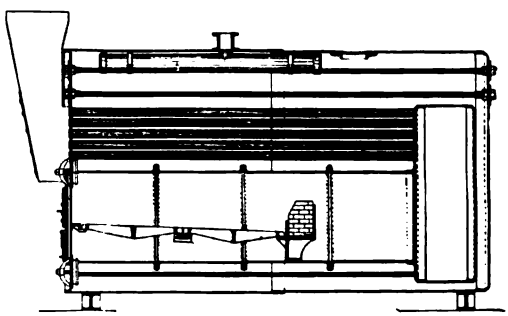


FIG. 210. FIG. 211.
END ELEVATION AND SECTION OF (SO CALLED) SCOTCH BOILER.

will practically soot up completely in time; while only a limited amount of soot will lodge on the water tube, whether it is looked after or not. (See Figs. 207 and 208.)

In fitting up a large ice factory with water tube boilers, and modern distilling plant, there would be no difficulty (with apparatus such as that described in Chapter XIX) in insuring the tubes being kept absolutely free from deposit, by first evaporating all the water supplied as feed. At sea special provision is now made for supplying the "make up" as it is termed, and one of the evaporators used would be applicable for smaller installations. Fig. 209 is an evaporator, so constructed that when the copper steam coils are coated up on the outside, by the salts of lime, magnesia or other mineral removed from the water, they can be swung right clear out of the casing for easy cleaning.

Internally fired boilers with return tubes have many advocates, and although primarily a marine type, they are much appreciated in many factories on land. In the United States they seem to have gotten the name of "Scotch" boilers—why is not very clear, as they are not so called in England or Australia. Figs. 210 and 211 represent two views of this type, designed for land use with a good chimney draft (for sea the proportion is generally much shorter in relation to the diameter). It is evident that with such boilers in an ice factory, near to brine tanks or cold rooms, their shells should be well covered with the best non-conducting composition, to prevent the radiation of heat.

Figs. 212 and 213 are sections of R. Munroe & Sons' safety and vertical water tube boilers, made at Pittsburg, Pa., U. S. A., and widely used in America.

All the plates used in the construction of this boiler are made of open hearth homogeneous flange steel. There are two water chambers made in exact duplicate of each other, the outer heads of which are dished. The outer head of the front chamber contains from two to six patented eclipse manholes according to the size of the boiler. These manheads permit of free access to all of the horizontal tubes. The thickness of the material used in these water chambers varies from five-sixteenths to seven-sixteenths, according to the size of the boiler. The chambers are made extra strong by being

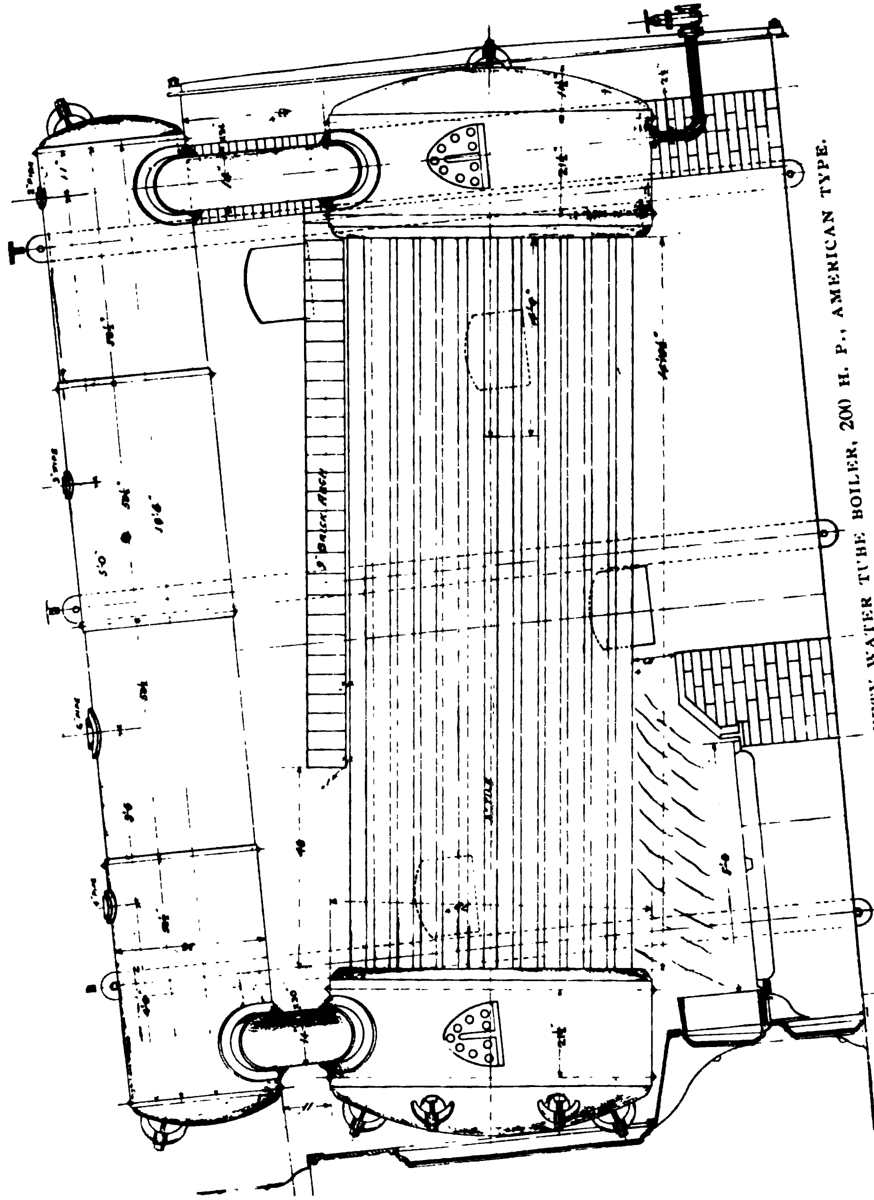


FIG. 212.—MUNROE'S SAFETY WATER TUBE BOILER, 200 H. P., AMERICAN TYPE.

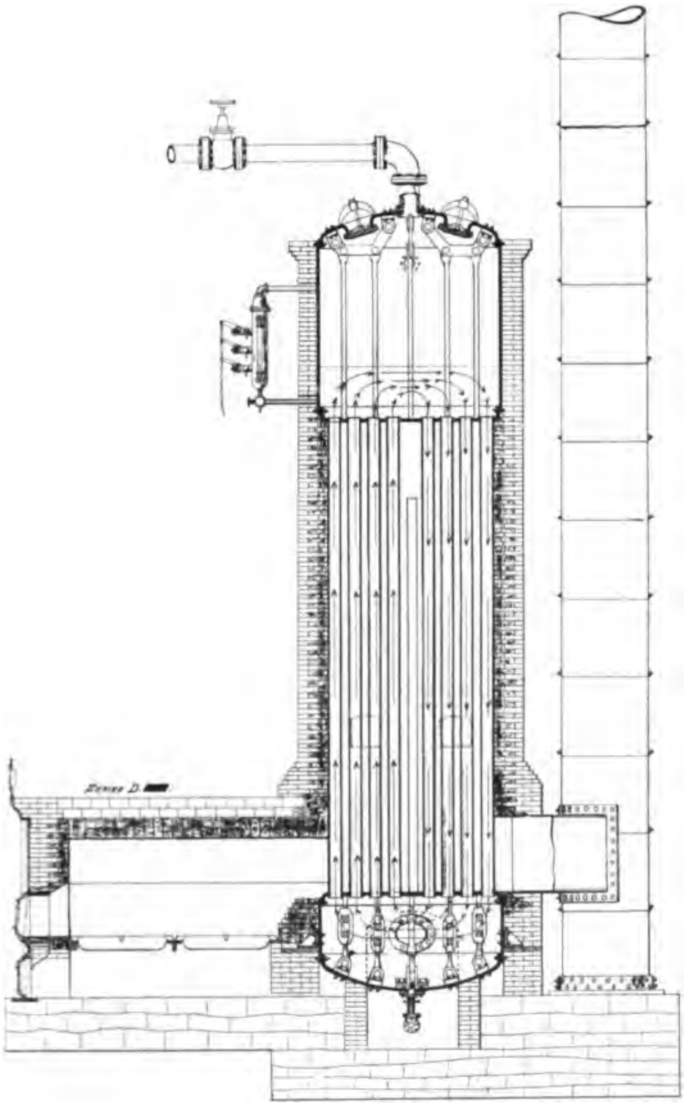


FIG. 213.—MUNROE'S VERTICAL WATER TUBE BOILER, AMERICAN TYPE.

double or triple staggered, riveted at the point where the sheets are lapped. The tube sheets of water chambers are five-eighths of an inch thick, and made of homogeneous flange steel. Riveted on to the outer heads of the water chambers are from four to eight angle braces, according to the size of the boiler, and connected to the braces are connecting or tension rods. These rods are made of soft iron, and they are connected to the braces by pins varying from three-fourths of an inch to one and one-quarter inches in diameter. The tension rods are from one inch to one and one-half inches in diameter; they have a swivel on one end, so that they can be easily removed at any time for cleaning or repairs.

The steam and water drum is made of homogeneous flange steel, same as the water chambers, the outer heads of the drum being dished, the front or rear head containing a manhole, placed in the head opposite to stack.

The water legs are made of the same material as the drum and chambers, and they are of ample size to meet the requirements of the various boilers, the front leg being larger than the rear, so as to allow a large liberating surface. The water legs are double riveted at their flanges to the water chambers and steam and water drums.

The horizontal tubes are four inches in diameter, and vary in length according to the size of the boiler. They are placed in a staggered position in the tube sheets and expanded with a Dudgeon expander, then turned over.

The boiler is set up on brick work and suspended by heavy cast iron lugs riveted at their proper angles on to the water chambers. Each boiler has four lugs, two on each water chamber.

Over the grate bars and under front water chamber is placed an arch, and immediately over the tubes there is another arch made of fire brick; it is built from side wall to side wall of the boiler, and it runs from the front water leg to a point, changing according to the size of the boiler, so as to allow a sufficient draft area. Resting on the top row of drop or circulating tubes is the tile arch to form a cover for the draft area; the other end of the arch resting on the top of the side walls. The pipes in side walls are of a diameter to admit sufficient oxygen to facilitate combustion. As the tubes

are on an angle of one and one-fourth inches to the foot, good results must be obtained, as the heat units impinge directly on the tubes, and as the tubes are staggered, the heat units are distributed over the entire heating surface. Ten square feet of heating surface is allowed per horse power.

The water is fed through a pipe which is connected to the steam and water drum at a point directly opposite the center of the rear water leg, and which extends almost to the center of the rear water chamber, and steam is taken out of the opposite end.

THE HOLDEN ICE MAKING SYSTEM.

Although nothing has heretofore been said as to the relative merits of the can and plate systems of ice making, as they form quite a separate question from those that have been discussed, .the opportunity may be here taken to describe an entirelv different system of ice making, which has been recently introduced by Mr. D. L. Holden, of Philadelphia, U. S. A., with what ultimate success remains to be seen.

Reference was made on pages 71 and 74 to the effect of velocity and thickness of material in its effect on the conduction of heat. The effect of the low conductivity of ice in reducing the ratio at which it forms, either in the can or on the plate, is well known and exercises a great influence in restricting the thickness of the blocks as made in actual practice. Mr. Holden makes a wide departure from the usual method (although his ideas are not new as applied on a small scale for ice cream freezing), and constructs his refrigerator as a hollow cylinder, which rotates in the water to be frozen. The liquid ammonia is carried in through one of the trunnions, and the other one is connected either to the re-absorber in the case of an absorption plant, or to the suction side of the compressor in a compression system; and the evaporation of the ammonia in this cylinder freezes the water in contact with its metallic surface very rapidly.

According to accounts appearing in contemporary journals, this rate of freezing is so fast that it would incrust the cylinder at the rate of a quarter of an inch per minute; but as soon as it is formed, it is cut or shaved off by a set of rotary

knives, and these ice shavings are carried by a creeper or conveyor to hydraulic presses. When inclosed by brass molds and under a pressure of between 300 and 400 pounds to the inch, this mush ice is so compressed that all the water and air is got rid of, and regelation is brought about. This regelation produces blocks of compact and solid ice, but whether by sufficient pressure crystal clear ice can be thus made is not stated. It is claimed, however, that ice can be made cheaper in this manner. If the process really turns out to be cheaper, then it must be so by the saving effected in the plant—labor and accessories connected with the actual ice making. It certainly cannot make more ice with a given amount of refrigerating effect as produced by a machine, than an ordinary refrigerating tank and metal molds can do, if there is careful insulation and absence of waste in thawing out.

IN CONCLUSION.

In taking leave of the reader the author would say that, in commencing this task he had no idea that his original paper would expand to the dimensions this work has now assumed, but he is now quite aware that there are sufficient interesting matters omitted in connection with the machinery of refrigeration to make another volume. He cannot let the opportunity pass without expressing his obligation to the publishers for the handsome dress they have put him into, and for their artistic reproduction of his original drawings.

To builders of refrigerating machinery who have kindly forwarded him catalogues and information he here expresses his obligation, and as gratitude is said to be "a lively sense of favors to come," he hopes to be the recipient of any new editions that may be published. To such builders or managers of refrigerating machinery as may be numbered among the readers of the ideas herein set forth, he would say that any information or suggestions in connection with that department of engineering, with which they may favor him, will be much appreciated and duly acknowledged.

I.
APPENDIX

TABLES.

TABLE SHOWING THE MEAN PRESSURE OF STEAM IN CYLINDERS OF COMPOUND AND TRIPLE-EXPANSION ENGINES.

WITH VARIOUS INITIAL STEAM PRESSURES, EXPANDING DOWN TO A NOMINAL TERMINAL PRESSURE OF 15 LBS. PER SQUARE INCH.

Ratio of Expansion, or number of times steam is expanded.	Hyperbolic Logarithms of the Ratio of Expansion.	Points of cut-off in fractions of the Stroke, reckoned from the commencement.	Mean pressure during the stroke, the initial pressure being taken as = 1.	ABSOLUTE PRESSURE. Initial pressure in lb. per square suitable for given ratio of expansion.	ABSOLUTE PRESSURE. Mean pressure in lb. per square inch.
6	1.7918	1/6	0.4653	90	41.8
6¼	1.8326	4/25	0.4532	93.75	42.4
6½	1.8718	2/13	0.4418	97.5	43
6¾	1.9095	4/27	0.4310	101.25	43.4
7	1.9459	1/7	0.4208	105	44
7¼	1.9810	4/29	0 4111	108.75	44.6
7½	2.0149	2/15	0.4002	112.5	45
7¾	2.0477	4/31	0.3932	116.25	45.6
8	2.0794	1/8	0.3849	120	46
8¼	2.1102	4/33	0.3779	122.75	46.3
8½	2.1401	2/17	0.3694	127.5	47
8¾	2.1691	4/35	0.3621	131.25	47.5
9	2.1972	1/9	0.3552	135	47.9
9¼	2.2246	4/37	0.3486	138.75	48.3
9½	2.2513	2/19	0.3122	142.5	48.7
9¾	2.2773	4/39	0.3361	146.25	49
10	2.3026	1/10	0.3302	150	49 5
10¼	2.3279	4/41	0.3246	153.75	49.8
10½	2.3513	2/21	0.3191	157.5	50.2
10¾	2.3749	4/43	0.3139	161.25	50.6
11	2.3979	1/11	0.3089	165	50.9
11¼	2.4201	4/45	0.3010	168.75	51.2
11½	2.4430	2/23	0.2993	172.5	51.6
11¾	2.4636	4/47	0.2947	176.25	51.9
12	2.4849	1/12	0.2904	180	52.2
12¼	2.5052	4/49	0.2861	183.75	52.3
12½	2.5262	2/25	0.2821	187.5	52.8
12¾	2.5455	4/51	0.2780	191.25	53
13	2.5649	1/13	0.2742	195	53.4
13¼	2.5840	4/53	0.2704	198.75	53.8
13½	2.6027	2/27	0.2668	202.5	54
13¾	2.6211	4/55	0.2633	206.25	54.2
14	2.6391	1/14	0.2599	210	54.5
14¼	2.6567	4/57	0.2566	213.75	54.8
14½	2.6740	2/29	0.2533	217.5	55
14¾	2.6913	4/59	0.2502	221.25	55.3
15	2.7081	1/15	0.2472	225	55.6
15½	2.7408	2/31	0.2412	232.5	56
16	2.7726	1/16	0.2358	240	56.5

MEAN SPEED OF COMPRESSOR AND ENGINE PISTONS

IN FEET PER MINUTE.

Stroke in inches.	REVOLUTIONS PER MINUTE.																									Stroke in Inches.
	50	60	75	80	90	100	120	125	140	150	160	170	180	200	225	240	250	275	300	320	325	350	360	375	400	
6	50	60	75	80	90	100	120	125	140	150	160	170	180	200	225	240	250	275	300	320	325	350	360	375	400	6
7	58	70	87	93	105	117	140	146	163	175	187	198	210	233	262	280	292	321	350	373	379	408	420	437	467	7
8	67	80	100	107	120	133	160	167	187	200	213	227	240	267	300	320	333	367	400	427	433	467	480	500	533	8
9	75	90	112	120	135	150	180	187	210	225	240	255	270	300	337	360	375	412	450	480	487	525	540	562	600	9
10	83	100	125	133	150	167	200	208	233	250	267	283	300	333	375	400	417	458	500	533	542	583	600	625	667	10
11	92	110	137	147	165	183	220	229	257	275	293	312	330	367	412	440	458	504	550	587	596	642	660	687	733	11
12	100	120	150	160	180	200	240	250	280	300	320	340	360	400	450	480	500	550	600	640	650	700	720	750	800	12
14	117	140	175	187	210	233	280	292	303	350	373	397	420	466	525	560	583	642	700	747	758	817	840	875	933	14
15	125	150	187	200	225	250	300	312	350	375	400	425	450	500	562	600	625	687	750	800	812	875	900	937	1000	15
16	133	160	200	213	240	267	320	333	373	400	427	453	480	533	600	640	667	733	800	853	867	933	960	1000	1067	16
18	150	180	225	240	270	300	360	375	420	450	480	510	540	600	675	720	750	825	900	960	975	1050	1080	1125	1200	18
20	167	200	250	267	300	334	400	417	467	500	533	587	600	667	750	800	833	917	1000	1067	1083	1167	1200	1250	1333	20
22	193	220	275	293	330	367	440	458	513	550	587	623	660	733	825	880	917	1008	1100	1173	1192	1233	1320	1375	1467	22
24	200	240	300	320	360	400	480	500	560	600	640	680	720	800	900	960	1000	1100	1200	1280	1300	1400	1440	1500	1600	24
26	217	260	325	347	390	433	520	542	607	650	693	733	780	867	975	1040	1083	1192	1300	1360	1408	1517	1560	1625	1736	26
28	233	280	350	373	420	467	560	583	653	700	746	793	840	933	1050	1120	1166	1283	1400	1494	1516	1634	1680	1750	1861	28
30	250	300	375	400	450	500	600	625	700	750	800	850	900	1000	1125	1200	1250	1375	1500	1600	1624	1750	1800	1874	2000	30
36	300	360	450	480	540	600	720	750	840	900	960	1020	1080	1200	1350	1440	1500	1650	1800	1920	1950	2100	2160	2250	2400	36
40	333	400	500	533	600	667	800	833	933	1000	1066	1133	1200	1333	1500	1600	1666	1834	2000	2134	2166	2334	2400	2500	2666	40
42	350	420	525	560	630	700	840	875	980	1050	1120	1190	1360	1400	1575	1680	1749	1926	2100	2241	2274	2451	2520	2625	2799	42
48	400	480	600	640	720	800	960	1000	1120	1200	1300	1260	1440	1600	1800	1920	2000	2200	2400	2560	2600	2800	2880	3000	3200	48
54	450	540	675	720	810	900	1080	1125	1260	1350	1440	1530	1620	1800	2025	2160	2250	2475	2700	2880	2925	3150	3250	3375	3600	54
60	500	600	750	800	900	1000	1200	1250	1400	1500	1600	1700	1800	2000	2250	2400	2500	2750	3000	3200	3248	3500	3600	3748	4000	60
66	550	660	825	880	990	1100	1320	1375	1540	1650	1761	1869	1980	2199	2475	2640	2751	3024	3300	3519	3576	3699	3960	4125	4401	66
72	600	720	900	960	1080	1200	1440	1500	1680	1800	1920	2040	2160	2400	2700	2880	3000	3300	3600	3840	3900	4200	4320	4500	4800	72

PERCENTAGE OF SAVING OF FUEL BY HEATING FEED WATER.

The following table gives the percentage of saving for a steam pressure of sixty pounds per square inch, with various initial and final temperatures of the feed:

Temperature of Water after Heating.	Initial Temperature of Water.																
	40°	45°	50°	55°	60°	65°	70°	75°	80°	85°	90°	95°	100°	110°	120°	130°	140°
110°	5.11	4.70	4.30	3.88	3.47	3.05	2.63	2.20	1.77	1.33	0.89	0.45	.....	.....	.....	.....	.....
110	5.96	5.56	5.15	4.74	4.33	3.92	3.50	3.07	2.65	2.22	1.78	1.34	0.90	.....	.....	.....	.....
120	6.81	6.41	6.01	5.60	5.20	4.79	4.37	3.95	3.53	3.10	2.66	2.23	1.79	0.90	.....	.....	.....
130	7.67	7.27	6.88	6.47	6.07	5.66	5.25	4.84	4.42	4.00	3.56	3.13	2.70	1.82	0.92	.....	.....
140	8.52	8.13	7.73	7.33	6.93	6.53	6.12	5.71	5.30	4.88	4.45	4.02	3.59	2.72	1.83	0.92	.....
150	9.38	8.99	8.60	8.20	7.81	7.41	7.00	6.60	6.19	5.77	5.34	4.92	4.50	3.63	2.75	1.85	0.94
160	10.23	9.85	9.46	9.07	8.68	8.29	7.89	7.48	7.07	6.66	6.24	5.82	5.40	4.55	3.68	2.78	1.88
170	11.08	10.70	10.32	9.93	9.55	9.15	8.76	8.36	7.95	7.55	7.13	6.71	6.30	5.45	4.59	3.70	2.01
180	11.94	11.57	11.19	10.80	10.42	10.03	9.64	9.24	8.84	8.44	8.02	7.62	7.20	6.36	5.51	4.63	3.74
190	12.80	12.43	12.05	11.67	11.29	10.91	10.52	10.13	9.73	9.33	8.92	8.52	8.11	7.28	6.43	5.56	4.68
200	13.66	13.29	12.92	12.54	12.17	11.79	11.40	11.01	10.62	10.23	9.82	9.42	9.01	8.19	7.35	6.49	5.62
210	14.53	14.16	13.79	13.42	13.05	12.67	12.29	11.91	11.52	11.13	10.72	10.32	9.93	9.11	8.28	7.43	6.57
212	14.69	14.33	13.96	13.59	13.22	12.85	12.47	12.08	11.69	11.30	10.90	10.51	10.10	9.29	8.46	7.61	6.75
220	15.38	15.02	14.66	14.29	13.92	13.55	13.17	12.79	12.41	12.02	11.62	11.23	10.83	10.02	9.20	8.36	7.51
230	16.24	15.89	15.52	15.16	14.79	14.43	14.05	13.68	13.30	12.91	12.52	12.13	11.74	10.99	10.12	9.29	8.44
240	17.10	16.75	16.39	16.03	15.67	15.30	14.93	14.56	14.18	13.81	13.42	13.03	12.64	11.85	11.04	10.22	9.38
250	17.97	17.62	17.26	16.91	16.55	16.19	15.82	15.45	15.08	14.71	14.32	13.94	13.55	12.77	11.97	11.16	10.33
260	18.83	18.49	18.14	17.78	17.43	17.07	16.71	16.35	15.98	15.61	15.22	14.85	14.47	13.69	12.91	12.10	11.28
270	19.69	19.35	19.01	18.66	18.31	17.95	17.59	17.23	16.87	16.50	16.12	15.75	15.37	14.61	13.83	13.03	12.22
280	20.55	20.21	19.87	19.52	19.18	18.83	18.47	18.12	17.76	17.39	17.02	16.65	16.12	15.52	14.75	13.96	13.16
290	21.42	21.08	20.75	20.40	20.06	19.71	19.36	19.01	18.65	18.30	17.92	17.56	17.19	16.44	15.68	14.90	14.10
300	22.28	21.95	21.61	21.27	20.93	20.59	20.25	19.90	19.54	19.19	18.82	18.46	18.09	17.36	16.60	15.82	15.04
310	23.14	22.82	22.49	22.15	21.82	21.48	21.14	20.79	20.44	20.09	19.73	19.37	19.01	18.28	17.53	16.76	15.99
320	24.00	23.68	23.35	23.02	22.69	22.35	22.02	21.67	21.33	20.98	20.62	20.27	19.91	19.19	18.45	17.69	16.93

WEIGHT OF CAST IRON PIPES IN POUNDS

PER LINEAL FOOT.

Internal Diameter.	Thickness of Metal.								
	¼ inch.	⅜ inch.	½ inch.	⅝ inch.	¾ inch.	⅞ inch.	1 inch.	1⅛ inch.	1¼ inch.
Inch.	Pounds.	Pounds.	Pounds.	Pounds.	Pounds.	Pounds.	Pounds.	Pounds.	Pounds.
1	3.06	5.06	.....	.....	.....	.....	.....	......	......
1¼	3.68	5.98	.....	.....	.....	.....	.....	......	......
1½	4.29	6.90	9.82	.....	.....	.....	.....	......	......
1¾	4.91	7.83	11.05	.....	.....	.....	.....	......	......
2	5.53	8.75	12.27	16.11	.....	.....	.....	.....	......
2¼	6.14	9.66	13.05	17.64	.....	.....	.....	......	......
2½	6.74	10.58	14.72	19.17	.....	.....	.....	......	......
2¾	7.36	11.50	15.95	20.70	.....	.....	.....	.....	......
3	7.98	12.43	17.18	22.19	27.62	.....	.....	......	......
3¼	8.59	13.34	18.35	23.78	29.45	.....	.....	......	......
3½	9.20	14.21	19.64	25.31	31.03	37.53	.....	......	......
3¾	9.76	15.19	20.86	26.85	33.13	39.73	.....	......	......
4	10.44	16.11	22.10	28.38	34.98	41.88	49.09	......	......
4¼	11.10	17.08	23.37	29.97	36.87	44.08	51.60	......	......
4½	11.66	17.94	24.54	31.44	38.65	46.17	53.99	62.12	......
4¾	12.27	18.89	25.77	32.98	40.50	48.32	56.45	64.89	......
5	12 88	19.78	26.99	34.51	42.33	50.46	58.90	67.64	76.69
5¼	13.50	20.71	28.23	36.05	44.18	52.62	61.36	70.41	79.77
5½	14.11	21.63	29.45	37.58	46.02	54.76	63.81	73.17	82.84
5¾	14.73	22.55	30.68	39.12	47.86	56.91	66.27	75.94	85.91
6	15.34	23.47	31.91	40.65	49.70	59.06	68.73	78.70	88.75
6¼	15.95	24.39	33.13	42.18	51.54	61.21	71.18	81.23	92.04
6½	16.57	25.31	34.36	43.72	53.39	63.36	73.41	84.22	95.10
6¾	17.18	26.23	35.59	45.26	55.23	65.28	76.09	86.97	98.18
7	17.79	27.15	36.82	46.79	56.84	67.65	78.53	89.74	101.24
7¼	18.41	28.08	38.05	48.10	58.91	69.79	81.00	92.50	104.31
7½	19.03	29.00	39.05	49.86	60.74	71.95	83.45	95.26	107.38
7¾	19.64	29.69	40.50	51.38	62.59	74.09	85.90	98.02	110.45
8	20.02	30.83	41.71	52.92	64.42	76.23	88.35	100.78	113.51
8¼	20.86	31.74	42.95	54.45	66.26	78.38	90.81	103.54	116.58
8½	21.69	32.90	44.40	56.21	68.33	80.76	93.49	106.53	119.87
8¾	22.09	33.59	45.40	57.52	69.95	82.68	95.72	109.06	122.72
9	22.71	34.52	46.64	59.07	71.80	84.84	98.18	111.84	125.80
9¼	.....	35.43	47.86	60.59	73.63	86.97	100.63	114.59	128.85
9½	.....	36.36	49.09	62.13	75.47	89.13	103.09	117.35	131.93
9¾	.....	37.28	50.32	63.66	77.32	91.28	105.54	120.12	134.99
10	....	38.20	51.54	65.20	79.16	93.42	108.00	122.87	138.06
10¼	.....	.....	52.77	66.73	80.99	95.57	110.44	125.63	141.12
10½	.....	.....	54.00	68.26	82.84	97.71	112.90	128.39	144.19
10¾	.....	.....	55.22	69.80	84.67	99.86	115.35	131.15	147.26
11	.....	.....	56.46	71.33	86.52	102.01	117.81	133.92	150.33
11¼	.....	....	.....	72.86	88.35	104.15	120.26	136.67	153.40
11½	.....	.....	.....	74.39	90.19	106.30	122.71	139.44	156.44
11¾	.....	.....	.....	75.93	92.04	108.45	125.18	142.18	159.54
12	.....	.....	.....	77.46	93.60	110.60	127.60	144.96	162.60

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USEFUL TABLES AND MEMORANDA RELATING TO PRIME MOVERS.

AREAS OF CIRCLES ADVANCING BY EIGHTHS OF AN INCH.

Diam.	0	1/8	1/4	3/8	1/2	5/8	3/4	7/8	Diam.
0	.0000	.0122	.0490	.1104	.1963	.3068	.4417	.6013	0
1	.7854	.9940	1.227	1.484	1.767	2.073	2.405	2.761	1
2	3.141	3.546	3.976	4.430	4.908	5.411	5.939	6.491	2
3	7.068	7.669	8.295	8.946	9.621	10.32	11.04	11.79	3
4	12.76	13.36	14.18	15.03	15.90	16.80	17.72	18.76	4
5	19.63	20.62	21.64	22.69	23.75	24.85	25.96	27.10	5
6	28.27	29.46	30.67	31.91	33.18	34.47	35.78	37.12	6
7	38.48	39.87	41.28	42.71	44.17	45.66	47.17	48.70	7
8	50.26	51.84	53.45	55.08	56.74	58.42	60.13	61.86	8
9	63.61	65.39	67.20	69.02	70.88	72.75	74.76	76.58	9
10	78.54	80.51	82.51	84.54	86.59	88.66	90.76	92.88	10
11	95.03	97.20	99.40	101.6	103.8	106.1	108.4	110.7	11
12	113.0	115.4	117.8	120.2	122.7	125.1	127.6	130.1	12
13	132.7	135.2	137.8	140.5	143.1	145.8	148.4	151.2	13
14	153.9	156.6	159.4	162.2	165.1	167.9	170.8	173.7	14
15	176.7	179.6	182.6	185.6	188.6	191.7	194.8	197.9	15
16	201.0	204.2	207.3	210.5	213.8	217.0	220.3	223.6	16
17	226.9	230.3	233.7	237.1	240.5	243.9	247.4	250.9	17
18	254.4	258.0	261.5	265.1	268.8	272.4	276.1	279.8	18
19	283.5	287.2	291.0	294.8	298.6	302.4	306.3	310.2	19
20	314.1	318.1	322.0	326.0	330.0	334.1	338.1	342.2	20
21	346.3	350.4	354.6	358.8	363.0	367.2	371.5	375.8	21
22	380.1	384.4	388.8	393.2	397.6	402.0	406.4	410.9	22
23	415.4	420.0	424.5	429.1	433.7	438.3	443.0	447.6	23
24	452.3	457.1	461.8	466.6	471.4	476.2	481.1	485.9	24
25	490.8	495.7	500.7	505.7	510.7	515.7	520.7	525.8	25
26	530.9	536.0	541.1	546.3	551.5	556.7	562.0	567.2	26
27	572.5	577.8	583.2	588.5	593.9	599.3	604.8	610.2	27
28	615.7	621.2	625.7	632.3	637.9	643.5	649.1	654.8	28
29	660.5	666.2	671.9	677.7	683.4	689.2	695.1	700.9	29
30	706.8	712.7	718.6	724.6	730.6	736.6	742.6	748.6	30
31	754.8	760.9	767.0	773.1	779.3	785.5	791.7	798.0	31
32	804.2	810.5	816.9	823.2	829.6	836.0	842.4	848.8	32
33	855.3	861.8	868.3	874.8	881.4	888.0	894.6	901.3	33
34	907.9	914.6	921.3	928.1	934.8	941.6	948.4	955.3	34
35	962.1	969.0	975.9	982.8	989.8	996.8	1003	1010	35
36	1017	1025	1032	1039	1046	1053	1060	1068	36
37	1075	1082	1089	1097	1104	1111	1119	1126	37
38	1134	1141	1149	1156	1164	1171	1179	1186	38
39	1194	1202	1210	1217	1225	1233	1241	1248	39
40	1256	1264	1272	1280	1288	1296	1304	1312	40
41	1320	1328	1336	1344	1352	1360	1369	1377	41
42	1385	1393	1402	1410	1418	1427	1435	1443	42
43	1452	1460	1469	1477	1486	1494	1503	1511	43
44	1520	1529	1537	1546	1555	1564	1572	1581	44
45	1590	1599	1608	1617	1626	1634	1643	1652	45
46	1661	1671	1680	1689	1698	1707	1716	1725	46
47	1734	1744	1753	1762	1772	1781	1790	1800	47
48	1809	1819	1828	1837	1847	1854	1868	1876	48
49	1885	1895	1905	1914	1924	1934	1943	1953	49
50	1963	1973	1983	1993	2003	2012	2022	2032	50

D=Diameter $D=\frac{C}{3.14159}$ or $\sqrt{A \div .7854}$ or $C \times .31831$.

A=Area. $A=D^2 \times .7854$ or $(C \div 3.5446)^2$.

C=Circumference. $C=D \times 3.14159$ or $3.5446 \sqrt{A}$.

S=Contents of Sphere $S=D^3 \times .5236$.

B=Contents of Cylinder. $B=A \times$ length. (A being the area of one end.)

USEFUL TABLES AND MEMORANDA RELATING TO PRIME MOVERS.

CIRCUMFERENCES OF CIRCLES ADVANCING BY EIGHTHS OF AN INCH.

Diam.	0	⅛	¼	⅜	½	⅝	¾	⅞	Diam.
0	.0	.3927	.7854	1.178	1.570	1.963	2.356	2.748	0
1	3.141	3.534	3.927	4.319	4.712	5.105	5.497	5.890	1
2	6.283	6.675	7.068	7.461	7.854	8.246	8.639	9.032	2
3	9.424	9.817	10.21	10.60	10.99	11.38	11.78	12.17	3
4	12.56	12.95	13.35	13.74	14.13	14.52	14.92	15.31	4
5	15.70	16.10	16.49	16.88	17.27	17.67	18.06	18.45	5
6	18.88	19.24	19.63	20.02	20.42	20.81	21.20	21.59	6
7	21.99	22.38	22.77	23.16	23.56	23.95	24.34	24.78	7
8	25.13	25.52	25.91	26.31	26.70	27.09	27.48	27.88	8
9	28.27	28.66	29.05	29.45	29.84	30.23	30.63	31.02	9
10	31.41	31.80	32.20	32.59	32.98	33.37	33.77	34.16	10
11	34.55	34.95	35.34	35.73	36.12	36.52	36.91	37.30	11
12	37.69	38.09	38.48	38.87	39.27	39.66	40.05	40.44	12
13	40.84	41.23	41.62	42.01	42.41	42.80	43.19	43.58	13
14	43.98	44.35	44.76	45.16	45.55	45.94	46.33	46.73	14
15	47.12	47.51	47.90	48.30	48.59	49.08	49.48	49.87	15
16	50.26	50.65	51.05	51.44	51.83	52.22	52.62	53.01	16
17	53.40	53.79	54.19	54.58	54.97	55.37	55.76	56.15	17
18	56.54	56.94	57.33	57.72	58.11	58.51	58.90	59.29	18
19	59.69	60.08	60.47	60.86	61.26	61.65	62.04	62.43	19
20	62.83	63.22	63.61	64.01	64.40	64.79	65.18	65.58	20
21	65.97	66.36	66.75	67.15	67.54	67.93	68.32	68.72	21
22	69.11	69.50	69.90	70.29	70.68	71.07	71.47	71.86	22
23	72.25	72.64	73.04	73.43	73.82	74.22	74.61	75.00	23
24	75.39	75.79	76.18	76.57	76.96	77.36	77.75	78.14	24
25	78.54	78.93	79.32	79.71	80.10	80.50	80.89	81.28	25
26	81.68	82.07	82.46	82 85	83.25	83.64	84.03	84.43	26
27	84.82	85.21	85.60	86.00	86.39	86.78	86.17	87.57	27
28	87.96	88.35	88.75	89.14	89.53	89.92	90.32	90.71	28
29	91.10	91.49	91.89	92.28	92.67	93.06	93.46	93.85	29
30	94.24	94.64	95.03	95.42	95.81	96.21	96.60	96.99	30
31	97.39	97.78	98.17	98.56	98.96	99.35	99.74	100.1	31
32	100.5	100.9	101.3	101.7	102.1	102.5	102.9	103.3	32
33	103.7	104.1	104.5	104.9	105.2	105.6	106.0	106.4	33
34	106.8	107.2	107.6	108.0	108.4	108.8	109.2	109.6	34
35	110.0	110.3	110.7	111.1	111.5	111.9	112.3	112.7	35
36	113.1	113.5	113.9	114.3	114.7	115.1	115.5	115.8	36
37	116.2	116.6	117.0	117.4	117.8	118.2	118.6	119.0	37
38	119.4	119.8	120.2	120.6	121.0	121.3	121.7	122.1	38
39	122.5	122.9	123.3	123.7	124.1	124.5	124.9	125.3	39
40	125.7	126.1	126.4	126.8	127.2	127.6	128.0	128.4	40
41	128.8	129.2	129.6	130.0	130.4	130.8	131.2	131.6	41
42	131.9	132.3	132.7	133.1	133.5	133.9	134.3	134.7	42
43	135.1	135.5	135.9	136.3	136.7	137.1	137.4	137.8	43
44	138.2	138.6	139.0	139.4	139.8	140.2	140.6	141.0	44
45	141.4	141.8	142.2	142.6	142.9	143.3	143.7	144.1	45
46	144.5	144.9	145.3	145.7	146.1	146.5	146.9	147.3	46
47	147.7	148.0	148.4	148.8	149.2	149.6	150.0	150.4	47
48	150.8	151.2	151.6	152.0	152.4	152.8	153.2	153.5	48
49	153.9	154.3	154.7	155.1	155.5	155.9	156.3	156.7	49
50	157.1	157.5	157.9	158.3	158.7	159.0	159.4	159.8	50

D=Diameter. $D = \frac{C}{3.14159}$ or $\sqrt{A \div .7854}$ or $C \times .31831$.

A=Area. $A = D^2 \times .7854$ or $(C \div 3.5446)^2$.

C=Circumference. $C = D \times 3.14159$ or $3.5446 \sqrt{A}$.

S=Contents of Sphere. $S = D^3 \times .5236$.

B=Contents of Cylinder. $B = A \times$ length. (A being the area of one end.)

TABLE OF COMPRESSOR CAPACITY IN CUBIC INCHES.

FROM 1 TO 36 INCHES DIAMETER OF CYLINDER, AND FROM 1 TO 24 INCHES STROKE.

The tabular number multiplied by strokes per minute and divided by 1,728 gives cubic feet per minute theoretical capacity of the cylinder.

Cylinder diam. in inches.	LENGTH OF STROKE IN INCHES.										Cylinder diam. in inches.
	1	2	3	4	5	6	7	8	9	10	
	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	
1	.785	1.571	2.356	3.141	3.927	4.712	5.498	6.283	7.068	7.854	1
1½	1.767	3.534	5.301	7.068	8.835	10.602	12.370	14.137	15.905	17.672	1½
2	3.141	6.283	9.425	12.566	15.705	18.849	21.991	25.132	28.274	31.416	2
2½	4.908	9.817	14.726	19.634	24.543	29.452	34.360	39.269	44.178	49.087	2½
3	7.068	14.137	21.206	28.274	35.343	42.411	49.480	56.549	63.617	70.686	3
4	12.566	25.132	37.698	50.265	62.830	75.396	87.962	100.53	113.09	125.66	4
5	19.635	39.270	58.905	78.540	98.175	117.81	137.44	157.08	176.71	196.35	5
6	28.274	56.548	84.822	113.09	141.37	169.64	197.92	226.19	254.46	282.74	6
7	38.484	76.968	115.45	153.93	192.42	230.90	269.39	307.87	346.35	384.84	7
8	50.265	100.53	150.79	201.06	251.32	301.59	351.85	402.12	452.38	502.65	8
9	63.617	127.23	190.85	254.47	318.08	381.70	445.32	508.93	572.55	636.17	9
10	78.540	157.08	235.62	314.16	392.70	471.24	549.78	628.32	706.86	785.40	10
11	95.033	190.06	285.09	380.13	475.16	570.19	665.23	760.26	855.29	950.33	11
12	113.09	226.18	339.27	452.36	565.45	678.54	791.63	904.72	1007.8	1120.9	12
13	132.73	265.46	398.19	530.92	663.65	796.38	929.11	1061.8	1194.5	1327.2	13
14	153.93	307.86	461.79	615.72	769.65	923.58	1077.5	1231.4	1385.3	1539.3	14
15	176.71	353.42	530.13	706.84	883.55	1060.2	1236.9	1413.6	1590.3	1767.1	15
16	201.06	402.12	603.18	804.24	1005.3	1206.3	1407.4	1608.4	1809.5	2010.6	16
17	226.98	453.96	680.94	907.92	1134.9	1361.8	1588.8	1815.8	2042.8	2269.8	17
18	254.46	508.92	763.38	1017.8	1272.3	1526.7	1781.2	2035.6	2290.1	2544.6	18
19	283.52	567.04	850.56	1134.0	1417.6	1701.1	1984.6	2268.1	2551.6	2835.2	19
20	314.16	628.32	942.48	1256.6	1570.8	1884.9	2199.1	2513.2	2827.4	3141.6	20
22	380.13	760.26	1140.4	1520.5	1900.6	2280.8	2660.9	3041.0	3421.1	3801.3	22
24	452.39	904.78	1357.1	1809.5	2261.9	2714.3	3166.7	3619.1	4071.5	4523.9	24
26	530.93	1061.8	1592.7	2123.7	2654.6	3185.5	3716.5	4247.4	4778.3	5309.3	26
28	615.75	1231.5	1847.2	2463.	3078.7	3694.5	4310.2	4926.	5541.7	6157.5	28
30	706.86	1413.7	2120.5	2827.4	3534.3	4241.1	4948.	5654.8	6361.7	7068.6	30
32	804.24	1608.4	2412.7	3216.9	4021.2	4825.4	5629.6	6433.9	7238.1	8042.4	32
34	907.92	1815.8	2723.7	3631.6	4539.6	5447.5	6355.4	7263.3	8171.2	9079.2	34
36	1017.8	2034.1	3051.2	4068.3	5085.4	6102.4	7119.5	8136.6	9153.7	1017.0	36

TABLE OF COMPRESSOR CAPACITY IN CUBIC INCHES.

FROM 1 TO 36 INCHES DIAMETER OF CYLINDER, AND FROM 1 TO 24 INCHES STROKE.

The tabular number multiplied by strokes per minute and divided by 1,728 gives cubic feet per minute theoretical capacity of the cylinder.

Cylinder diam. in inches.	LENGTH OF STROKE IN INCHES.										Cylinder diam. in inches.
	11	12	13	14	15	16	18	20	22	24	
	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	C. Ins.	
1	8.639	9.425	10.110	10.995	11.781	12.566	14.137	15.708	17.279	18.849	1
1½	9.439	21.206	22.973	24.740	26.507	28.274	31.809	35.343	38.877	42.411	1½
2	34.557	37.699	40.841	43.982	47.124	50.265	56.549	62.832	69.113	75.399	2
2½	53.995	58.904	63.813	68.721	73.630	78.539	88.356	98.174	107.99	117.81	2½
3	77.754	84.823	91.892	98.960	106.03	113.09	127.23	141.37	155.51	169.64	3
4	138.22	150.79	163.36	175.92	188.49	201.05	226.19	251.32	276.55	301.58	4
5	215.98	235.62	255.25	274.89	294.52	314.16	353.43	392.70	431.97	471.24	5
6	311.01	339.29	367.56	395.83	424.11	452.38	508.93	565.48	622.03	678.57	6
7	423.32	461.81	500.29	538.77	577.26	615.74	692.71	769.68	846.65	923.61	7
8	552.91	603.18	653.44	703.71	753.97	804.24	904.77	1005.3	1105.8	1206.3	8
9	699.78	763.40	827.02	890.63	954.25	1017.8	1145.1	1272.3	1399.5	1526.8	9
10	863.94	942.48	1021.0	1099.5	1178.1	1256.6	1413.7	1570.8	1727.8	1884.9	10
11	1045.3	1140.3	1235.4	1330.4	1425.4	1520.5	1710.5	1900.6	2090.7	2280.7	11
12	1233.9	1357.1	1470.2	1583.3	1696.4	1789.5	2035.7	2261.9	2488.1	2714.3	12
13	1459.9	1592.7	1725.5	1858.2	1980.9	2123.7	2389.1	2654.6	2920.1	3185.5	13
14	1693.2	1847.2	2001.1	2155.1	2309.0	2463.0	2770.8	3078.7	3386.6	3694.5	14
15	1943.8	2120.5	2297.2	2474.0	2650.7	2827.4	3180.8	3534.3	3887.7	4241.1	15
16	2211.6	2412.7	2613.8	2814.8	3015.9	3216.9	3619.1	4021.2	4423.3	4825.4	16
17	2496.8	2723.7	2650.7	3177.7	3404.7	3631.6	4085.6	4539.6	4993.5	5447.5	17
18	2799.0	3053.6	3308.0	3562.5	3817.0	4071.5	4580.4	5089.3	5598.3	6107.2	18
19	3118.7	3402.3	3685.8	3969.4	4252.9	4536.4	5103.5	5670.5	6237.6	6804.7	19
20	3455.7	3769.9	4084.0	4398.2	4712.4	5026.5	5654.8	6283.2	6911.5	7539.8	20
22	4181.4	4561.5	4941.7	5321.8	5701.9	6082.1	6842.3	7602.6	8362.9	9123.1	22
24	4976.2	5428.6	5881.0	6333.4	6785.8	7238.2	8143.0	9047.8	9952.5	10857	24
26	5840.2	6371.1	6902.0	7433.0	7963.9	8494.8	9556.7	10618	11680	12742	26
28	6773.2	7389.0	8004.7	8620.5	9236.3	9852.0	11083	12315	13546	14778	28
30	7775.4	8482.3	9189.1	9896.0	10602	11309	12723	14137	15550	16964	30
32	8846.6	9650.9	10455	11259	12063	12868	14576	16085	17693	19301	32
34	9987.1	10895	11802	12710	13618	14526	13642	18158	19974	21790	34
36	11187	12214	13232	14240	15258	16275	18311	20347	22383	24418	36

CORRECTION FOR TEMPERATURE OF AQUA AMMONIA.

THE FIGURES IN TOP ROW INDICATE DEGREES FAHRENHEIT; THOSE IN THE COLUMNS BENEATH GIVE THE STRENGTH OF AMMONIA AT 60°.

BY PERMISSION HENRY VOGT MACHINE CO.

Degrees Baume.	60°	66°	67°	68°	72°	74°	76°	78°	81°	84°	88°	90°	92°	95°	96°	100°	102°	108°	109°
18	18	....	....	17¾	....	....	17½	....	....	17¼	....	....	17	....	....	16¾	....	16½	....
18¼	18¼	....	....	18	....	....	17¾	....	....	17½	....	....	17¼	....	....	17	....	16¾	....
18½	18½	....	....	18¼	....	....	18	....	....	17¾	....	....	17½	....	....	17¼	....	17	....
18¾	18¾	....	....	18½	....	....	18¼	....	....	18	....	....	17¾	....	....	17½	....	17¼	....
19	19	....	....	18¾	....	....	18½	....	....	18¼	....	....	18	....	....	17¾	....	17½	....
19¼	19¼	....	....	19	....	....	18¾	....	....	18½	....	....	18¼	....	....	18	....	17¾	....
19½	19½	....	....	19¼	....	....	19	....	....	18¾	....	....	18½	....	....	18¼	....	18	....
19¾	19¾	....	....	19½	....	....	19¼	....	....	19	....	....	18¾	....	....	18½	....	18¼	....
20	20	....	19¾	....	....	19½	....	....	19¼	....	19	....	....	18¾	....	....	18½	....	18¼
20¼	20¼	....	20	....	....	19¾	....	....	19½	....	19¼	....	....	19	....	....	18¾	....	18½
20½	20½	....	20¼	....	....	20	....	....	19¾	....	19½	....	....	19¼	....	....	19	....	18¾
20¾	20¾	....	20½	....	....	20¼	....	....	20	....	19¾	....	....	19½	....	....	19¼	....	19
21	21	....	20¾	....	....	20½	....	....	20¼	....	20	....	....	19¾	....	....	19½	....	19¼
21¼	21¼	....	21	....	....	20¾	....	....	20½	....	20¼	....	....	20	....	....	19¾	....	19½
21½	21½	....	21¼	....	....	21	....	....	20¾	....	20½	....	....	20¼	....	....	20	....	19¾
21¾	21¾	....	21½	....	....	21¼	....	....	21	....	20¾	....	....	20½	....	....	20¼	....	20
22	22	21¾	....	....	21½	....	....	21¼	....	21	....	20¾	....	....	20½	....	20¼	20	....
22¼	22¼	22	....	....	21¾	....	....	21½	....	21¼	....	21	....	....	20¾	....	20½	20¼	....
22½	22½	22¼	....	....	22	....	....	21¾	....	21½	....	21¼	....	....	21	....	20¾	20½	....
22¾	22¾	22½	....	....	22¼	....	....	22	....	21¾	....	21½	....	....	21¼	....	21	20¾	....
23	23	22¾	....	....	22½	....	....	22¼	....	22	....	21¾	....	....	21½	....	21¼	21	....
23¼	23¼	23	....	....	22¾	....	....	22½	....	22¼	....	22	....	....	21¾	....	21½	21¼	....
23½	23½	23¼	....	....	23	....	....	22¾	....	22½	....	22¼	....	....	22	....	21¾	21½	....

CORRECTION FOR TEMPERATURE OF AQUA AMMONIA.—*Concluded.*

THE FIGURES IN TOP ROW INDICATE DEGREES FAHRENHEIT; THOSE IN THE COLUMNS BENEATH GIVE THE STRENGTH OF AMMONIA AT 60°.

BY PERMISSION HENRY VOGT MACHINE CO.

Degrees Baume.	60°	64°	65°	68°	70°	72°	75°	76°	80°	84°	85°	88°	90°	92°	95°	96°	100°	104°	105°
23¾	23¾	....	23½	....	23¼	....	23	....	22¾	....	22½	....	22¼	....	22	....	21¾	....	21½
24	24	....	23¾	....	23½	....	23¼	....	23	....	22¾	....	22½	....	22¼	....	22	....	21¾
24¼	24¼	....	24	....	23¾	....	23½	....	23¼	....	23	....	22¾	....	22½	....	22¼	....	22
24½	24½	....	24¼	....	24	....	23¾	....	23½	....	23¼	....	23	....	22¾	....	22½	....	22¼
24¾	24¾	....	24½	....	24¼	....	24	....	23¾	....	23½	....	23¼	....	23	....	22¾	....	22½
25	25	....	24¾	....	24½	....	24¼	....	24	....	23¾	....	23½	....	23¼	....	23	....	22¾
25¼	25¼	....	25	....	24¾	....	24½	....	24¼	....	24	....	23¾	....	23½	....	23¼	....	23
25½	25½	25¼	....	25	....	24¾	....	24½	24¼	24	....	23¾	....	23½	....	23¼	23	22¾	....
25¾	25¾	25½	....	25¼	....	25	....	24¾	24½	24¼	....	24	....	23¾	....	23½	23¼	23	....
26	26	25¾	....	25½	....	25¼	....	25	24¾	24½	....	24¼	....	24	....	23¾	23½	23¼	....
26¼	26¼	26	....	25¾	....	25½	....	25¼	25	24¾	....	24½	....	24¼	....	24	23¾	23½	....
26½	26½	26¼	....	26	....	25¾	....	25½	25¼	25	....	24¾	....	24½	....	24¼	24	23¾	....
26¾	26¾	26½	....	26¼	....	26	....	25¾	25½	25¼	....	25	....	24¾	....	24½	24¼	24	....
27	27	26¾	....	26½	....	26¼	....	26	25¾	25½	....	25¼	....	25	....	24¾	24½	24¼	....
27¼	27¼	27	....	26¾	....	26½	....	26¼	26	25¾	....	25½	....	25¼	....	25	24¾	24½	....
27½	27½	27¼	....	27	....	26¾	....	26½	26¼	26	....	25¾	....	25½	....	25¼	25	24¾	....
27¾	27¾	27½	....	27¼	....	27	....	26¾	26½	26¼	....	26	....	25¾	....	25½	25¼	25	....
28	28	27¾	....	27½	....	27¼	....	27	26¾	26½	....	26¼	....	26	....	25¾	25½	25¼	....

The specific gravity of aqua ammonia changes with the temperature at which it is measured with the hydrometer. The readings are too high if the temperature of the ammonia is over 60° F., and too low if under. In order to ascertain the exact strength of ammonia at 60° F. make the corrections for temperature in accordance with the table above, thus: 26½° ammonia, measured at a temperature of 80° F., is equal to 25¼° ammonia at a temperature of 60° F.

TABLE OF GAUGES.

Gauge Number.	English Imperial Legal Standard.	Birmingham or Stubbs, or "English Standard."	Birmingham, for Sheets not Iron or Steel.	Birmingham, for Iron Sheets.	Lancashire, one of Holtzapffels.	Warrington or Rylands.	Old English for Brass, etc.	Needle Wire.	Music Wire in England.	Whitworth's English Standard.	American New Legal Standard.	Brown & Sharpe American Standard.
0000000	.500	....	....	......	....	.500	......	....	......	....	.500	......
000000	.464	....	....	......	....	.468+	......	....	......	....	.468+	......
00000	.432	....	....	......	....	.437+	......	....	......	....	.437+	......
0000	.400	.454	....	......	....	.406+	......	....	......	....	.406	.460
000	.372	.425	....	......	....	.375	......	....	......	....	.375	.409+
00	.348	.380	....	......	....	.343+	......	....	......	....	.343+	.364+
0	.324	.340	....	......	....	.326	......	....	......	....	.312+	.324+
1	.300	300	.004	.312+	.227	.300	......	.045	......	.001	.281+	.289+
2	.276	.284	.005	.281+	.219	.274	......	.042	......	.002	.265+	.257+
3	.252	.259	.008	.250	.209	.250	......	.035	......	.003	.250	.229+
4	.232	.238	.010	.234+	.204	.229	......	.032	......	.004	.234+	.204+
5	.212	.220	.012	.218+	.201	.209	......	.028	......	.005	.218+	.181+
6	.192	.203	.013	.203+	.198	.191	......	.025	.018	.006	.203+	.162+
7	.176	.180	.015	.187+	.195	.174	......	.022	.019	.007	.187+	.144+
8	.160	.165	.016	.171+	.192	.159	......	.020	.020	.008	.171+	.128+
9	.144	.148	.019	.156+	.191	.146	......	.018	.021	.009	.156+	.114+
10	.128	.134	.024	.140+	.190	.133	......	.016	.022	.010	.140+	.101+
11	.116	.120	.029	.125	.189	.117	......	.014	.023	.011	.125	.090+
12	.104	.109	.034	.112+	.185	.100	......	.013	.025	.012	.109+	.080+
13	.092	.095	.036	.100	.180	.090	......	.012	.026+	.013	.093+	.071+
14	.080	.083	.041	.087+	.177	.079	.083	.010	.028	.014	.078+	.064+
15	.072	.072	.047	.075	.175	.069	.072	.009	.030	.015	.080+	.057+
16	.064	.065	.051	.062+	.174	.062+	.065	.008	.032	.016	.062+	.050+
17	.056	.058	.057	.056+	.169	.053	.058	.007	.033+	.017	.056+	.045+
18	.048	.049	.061	.050	.167	.047	.049	.005	.035	.018	.050	.040+
19	.040	.042	.064	.043+	.164	.041	.040	.004	.038	.019	.043+	.035+
20	036	.035	.067	.037+	.160	.036	.035	.003	.042	.020	.037+	.031+
21	.032	.032	.072	.034+	.157	.031+	.031+	.002	....	....	.034+	.028+
22	.028	.028	.074	.031+	.152	.028	.029+	....	......	.022	.031+	.025+
23	.024	.025	.077	.028+	.150	......	.027	....	......	....	.028+	.022+
24	.022	.022	.082	.025	.148	......	.025	....	......	.024	.025	.020+
25	.020	.020	.095	.023+	.146	......	.023	....	......	....	.021+	.017+
26	.018	.018	.103	.021+	.143	......	.020+	....	......	.026	.018+	.015+
27	.016+	.016	.113	.020+	.141	......	.018+	....	......	....	.017+	.014+
28	.014+	.014	.120	.018+	.138	......	016+	....	......	.028	.015+	.012+
29	.013+	.013	114	.017+	.134	......	.015+	....	......	....	.014+	.011+
30	.012+	.012	.126	.015+	.125	......	.013+	....	......	.030	.012+	.010+
31	.011+	.010	.133	.014+	.118	......	.012+	....	......	....	.010+	.008+
32	.010+	.009	.143	.012+	.115	......	.011+	....	......	.032	.010+	.007+
33	.010	.008	.145	......	.111	......	.010+	....	......	....	.009+	.007+
34	.009+	.007	.148	......	.109	......	.009+	....	......	.034	.008+	.006+
35	.008+	.005	.158	......	.107	......	.009	....	......	....	.007+	.005+
36	.007+	.004	.169	......	.105	......	.007+	....	......	.036	.007+	.005
37	.006+	....	....	......	.102	......	006+	....	......	....	.006+	.004+
38	.006	....	....	......	.100	......	.005+	....	......	.038	.006+	.003+
39	.005+	....	....	......	.098	......	.005	....	......	....	......	.002+
40	.004+	....	....	......	.096	......	.004+	....	......	.040	......	.003+
41	.004+	....	....	......	.095	......	......	....	......	....	......	......
42	.004	....	....	......	.091	......	......	....	......	....	......	......

MECHANICAL AND ELECTRICAL UNIT EQUIVALENTS.

Units.	Equivalent Value in Other Units.
1 heat unit =	1,048 watt seconds. 772 ft.-lb. 0.252 caloric (kg.d.). 108 kilogrammeters. 0.000291 kilowatt hour. 0.000388 H. P. hour. 0.0000667 lb. coal oxidized. 0.00087 lb. water evaporated at 212° F.
1 heat unit per square foot per min. =	0.021 watt per square inch. 0.0174 kilowatt. 0.0232 H. P.
1 foot-pound =	1.36 joules. 0.1383 kilogrammeter. 0.000000377 kilowatt hour. 0.000291 heat unit. 0.0000005 H. P. hour.
1 pound water evaporated at 212° F. =	0.33 kilowatt hour. 0.44 H. P. hour. 1,148 heat units. 124,200 kilogrammeters. 1,219,000 joules. 887,800 ft.-lbs.
1 H. P. =	746 watts. 0.746 kilowatts. 33,000 ft.-lbs. per minute. 550 ft.-lbs. per second. 2,580 heat units per hour. 43 heat units per minute. 0.71 heat unit per second. 2.25 lbs. water evaporated per hour at 212° F.
1 H. P. hour =	0.746 kilowatt hour. 1,980,000 ft.-lbs. 2,580 heat units. 273,740 kilogrammeters. 2.25 lbs. water evaporated at 212° F. 17.2 lbs. water raised from 62° to 212° F.

Units.	Equivalent Value in Other Units.
1 watt =	1 joule per second. 0.00134 H. P. 0.001 kilowatt. 3.44 heat units per hour. 0.73 ft.-lb. per second. 0.003 lb. of water evaporated per hour. 44.24 ft.-lbs. per minute.
1 kilowatt =	1,000 watts. 1.34 H. P. 2,656,400 ft.-lbs. per hour. 4,424 ft.-lbs. per. min. 73.73 ft.-lbs. per second. 3,440 heat units per hour. 573 heat units per minute. 9.55 heat units per second. 3 lbs. water evaporated per hour at 212° F.
1 kilowatt hour =	1,000 watt hours. 1.34 H. P. hours. 2,656,400 ft.-lbs. 3,600,000 joules. 3,440 heat units. 366,848 kilogrammeters. 3 lbs. water evaporated at 212° F. 22.9 lbs. water raised from 62° to 212° F.
1 kilogrammeter =	7.23 ft.-lbs. 0.00000366 H. P. hour. 0.00000272 kilowatt hour. 0.0092 heat unit.
1 joule =	1 watt second. 0.00000278 kilowatt hour. 0.102 kilogrammeter. 0.00094 heat unit. 0.73 ft.-lb.

TABLE OF CONVERSION FACTORS.

ENGLISH TO METRICAL.

Pounds per lineal foot.....	×	1.488	= kilos. per lineal metre.
Pounds per lineal yard....	×	0.496	= kilos. per lineal metre.
Tons per lineal foot.......	×	3333.323	= kilos. per lineal metre.
Tons per lineal yard......	×	1111.11	= kilos. per lineal metre.
Pounds per mile...........	×	0.2818	= kilos. per lineal metre.
Pounds per square inch...	×	0.0703	= kilos. per square centimetre.
Tons per square inch......	×	1.575	= kilos. per square millimetre.
Pounds per square foot....	×	4.883	= kilos. per square metre.
Tons per square foot......	×	10.936	= tonnes per square metre.
Tons per square yard.....	×	1.215	= tonnes per square metre.
Pounds per cubic yard....	×	0.5933	= kilos. per cubic metre.
Pounds per cubic foot.....	×	16.020	= kilos. per cubic metre.
Tons per cubic yard.......	×	1.329	= tonnes per cubic metre.
Grains per gallon.........	×	0.01426	= grammes per litre.
Pounds per gallon.........	×	0.09983	= kilos. per litre.
Gallons per square foot....	×	48.905	= litres per square metre.
Foot-pounds...............	×	0.1382	= kilogrammetres.
Foot-tons..................	×	0.3333	= tonne-metres.
Horse power..............	×	1.0139	= force de cheval.
Pounds per H. P..........	×	0.477	= kilos. per cheval.
Square feet per H. P......	×	0.0196	= square metre per cheval.
Cubic feet per H. P.......	×	0.0279	= cubic metre per cheval.
Heat units................	×	0.252	= calories.
Heat units per square foot.	×	2.713	= calories per square metre.

METRICAL TO ENGLISH.

Kilos. per lineal metre.........	×	0.672	= pounds per lineal foot.
Kilos. per lineal metre.........	×	2.016	= pounds per lineal yard.
Kilos. per lineal metre.........	×	0.0003	= tons per lineal foot.
Kilos. per lineal metre.........	×	0.0009	= tons per lineal yard.
Kilos. per lineal metre.........	×	3.548	= pounds per mile.
Kilos. per square centimetre...	×	14.223	= pounds per square inch.
Kilos. per square millimetre...	×	0.635	= tons per square inch.
Kilos. per square metre........	×	0.2048	= pounds per square foot.
Tonnes per square metre.......	×	0.0914	= tons per square foot.
Tonnes per square metre.......	×	0.823	= tons per square yard.
Kilos. per cubic metre.........	×	1.686	= pounds per cubic yard.
Kilos. per cubic metre.........	×	0.0624	= pounds per cubic foot.
Tonnes per cubic metre........	×	0.752	= tons per cubic yard.
Grammes per litre.............	×	73.09	= grains per gallon (Imperial).
Kilos. per litre.................	×	10.438	= pounds per gallon (Imperial).
Litres per square metre........	×	0.0204	= gallons per square foot.
Kilogrammetres................	×	7.233	= foot-pounds.
Tonne-metres..................	×	3.000	= foot-tons.
Force de cheval................	×	0.9863	= horse power.
Kilos. per cheval...............	×	2.235	= pounds per H. P.
Square metre per cheval........	×	10.913	= square feet per H. P.
Cubic metre per cheval.........	×	35.806	= cubic feet per H. P.
Calories.......................	×	3.968	= heat units.
Calories per square metre......	×	0.369	= heat units per square foot.

TABLE SHOWING THE COMPARATIVE PROPERTIES OF THE THREE PRINCIPAL GASES USED FOR REFRIGERATION, AND THE ORIGINAL AUTHORITIES.

Heat units expressed in British Thermal Units per pound of the respective gas.

Temperature of ebullition in deg. F. t	Absolute pressure in lbs. per sq. in. $P \div 144$	Total heat reckoned from 32° Fah. λ	Heat of liq'd reckoned from 32° Fah. q	Latent heat of evaporation. r	Heat equival't of extern'l work. APu	Internal latent heat ρ	Increase of volume during evapor'n. u	Density of vapor or weight of one cubic ft.
Deg. Fah.	Lbs.	B. T. U.	B. T. U.	B. T. U.	B. T. U.	B. T. U.	Cu. Ft.	Lbs.
		FOR SATURATED SULPHUR DIOXIDE GAS.						Ledoux.
—22	5.56	157.43	—19.56	176.99	13.59	163.39	13.17	.076
—13	7.23	158.64	—16.30	174.95	13.83	161.12	10.27	.097
— 4	9.27	159.84	—13.05	172.89	14.05	158.84	8.12	.123
5	11.76	161.03	— 9.79	170.82	14.26	156.56	6.50	.153
14	14.74	162.20	— 6.53	168.73	14.46	154.27	5.25	.190
23	18.31	163.36	— 3.27	166.63	14.66	151.97	4.29	.232
32	22.53	164.51	0.00	164.51	14.84	149.68	3.54	.282
41	27.48	165.65	3.27	162.38	15.01	147.37	2.93	.340
50	33.25	166.78	6.55	160.23	15.17	145.06	2.45	.407
59	39.93	167.90	9.83	158.07	15.32	142.75	2.07	.483
68	47.61	168.99	13.11	155.89	15.46	140.43	1.75	.570
77	56.39	170.09	16.39	153.70	15.59	138.11	1.49	.669
86	66.36	171.17	19.69	151.49	15.71	135.78	1.27	.780
95	77.64	172.24	22.98	149.26	15.82	133.45	1.09	.906
104	90.31	173.30	26.28	147.02	15.91	131.11	.91	1.046
		FOR SATURATED AMMONIA GAS.						Zeuner.
—40	10.22	538.65	—60.82*	599.47	550.69	48.77	25.61	.039
—31	13.23	542.34	—54.54	596.88	547.33	49.56	20.10	.050
—22	16.95	545.96	—47.88	593.87	543.53	50.32	15.93	.063
—13	21.51	549.54	—40.84	590.38	539.32	51.06	12.74	.078
— 4	27.04	553.08	—33.43	586.51	534.72	51.78	10.28	.097
5	33.67	556.57	—25.64	582.21	529.74	52.48	8.37	.119
14	41.58	560.00	—17.47	577.47	524.32	53.16	6.86	.145
23	50.91	563.41	— 8.92	572.33	518.53	53.81	5.67	.175
32	61.85	566.75	0.00	566.75	512.30	54.45	4.73	.210
41	74.55	570.06	9.30	560.76	505.69	55.06	3.96	.251
50	89.21	573.31	18.98	554.33	498.67	55.65	3.35	.296
59	105.99	576.52	29.03	547.49	491.26	56.23	2.85	.348
68	125.08	579.68	39.46	540.22	483.44	56.78	2.44	.406
77	146.64	582.78	50.27	532.51	475.20	57.31	2.10	.471
86	170.83	585.84	61.45	524.39	466.58	57.81	1.82	.543
95	197.83	588.86	73.02	515.84	457.54	58.30	1.58	.622
104	227.76	591.83	84.95	506.88	448.11	58.77	1.39	.709
		SATURATED CARBONIC ACID GAS.						Pf. Schroter
—22	210	98.35	—37.80	136.15	16.20	119.95	.4138	2.321
—13	249	99.14	—32.51	131.65	16.04	115.61	.3459	2.759
— 4	292	99.88	—26.91	126.79	15.80	110.99	.2901	3.265
5	342	100.58	—20.92	121.50	15.50	106.00	.2435	3.853
14	396	101.21	—14.49	115.70	15.08	100.62	.2042	4.535
23	457	101.81	— 7.56	109.37	14.58	94.79	.1711	5.331
32	525	102.35	0.00	102.35	13.93	88.42	.1426	6.265
41	599	102.84	8.32	94.52	13.14	81.38	.1177	7.374
50	680	103.24	17.60	85.64	12.15	73.49	.0960	8.708
59	768	103.59	28.22	75.37	10.91	64.46	.0763	10.356
68	864	103.84	40.86	62.98	9.29	53.69	.0577	12.480
77	968	103.95	57.06	46.89	7.06	39.83	.0391	15.475
86	1080	103.72	84.44	19.28	2.95	16.33	.0147	21.519

HEAT OF COMBUSTION OF FUELS.

FUELS.	Air Chemically Consumed per Pound of Fuel.		Total Heat of Combustion of One Pound of Fuel.	Equivalent Evaporative Power from and at 212° F. Water per Pound of Fuel.
	Pounds	Cubic Feet at 62° F.	Units.	Pounds.
Coal of average composition.	10.7	140	14,700	15.22
Coke	10.81	142	13,548	14.02
Lignite	8.85	116	13,108	13.57
Asphalt	11.85	156	17,040	17.64
Wood, desiccated	6.09	80	10,974	11.36
Wood, 20 per cent moisture.	4.57	60	7,951	8.20
Wood charcoal, desiccated.	9.51	125	13,006	13.46
Peat, desiccated	7.52	99	12,279	12.71
Peat, 30 per cent moisture.	5.24	69	8,260	9.53
Straw	4.26	56	8,144	8.43
Petroleum	14.33	188	20,411	21.13
Petroleum oils	17.93	235	27,531	28.50
Coal gas per cubic foot at 62° F.	.....	...	630	0.70

In practice it is found that from eighteen to twenty-four pounds of air is required for the combustion of each pound of coal, according as to whether forced or natural draft is used.

COMPARATIVE EVAPORATIVE VALUE OF FUELS.

The feed water being 212° Fahrenheit when it enters the boiler, the following results were obtained from the consumption of one pound of the under-mentioned fuels. The first eight give the average of many samples tested by Messrs. Delabèche and Playfair:

FUELS.	Specific Gravity.	Pounds of Water Evaporated.		Comparative Values.
Welsh coal	1.315	9.05	By trial.	1.000
Newcastle coal	1.256	8.01		0.885
Derby and York coal	1.292	7.58		0.837
Lancashire coal	1.273	7.94		0.877
Scotch coal	1.260	7.70		0.851
British average	1.290	8.13		0.898
Irish anthracite	1.590	9.85		1.088
Patent fuels	1.167	9.20		1.016
French coal (average)	1.310	8.00	Approx.	0.884
Lignites (average)	1.198	6.66		0.736
Well dried peat	1.300	4.52		0.500
Coke (average)	0.750	9.00		0.995
Oak	0.930	4.52		0.500
Pine	0.660	2.5		0.276

AVERAGE COMPOSITION OF FUELS.

FUELS.	Carbon.	Hydrogen.	Sulphur.	Nitrogen.	Oxygen.	Ash.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
British coal...	80	5	1.25	1½	8	4
Coke..........	93½	..	1	..	..	5.5
Lignite.......	69	5	..	20		6
Asphalt.......	79	9	..	9		3
Wood, dry	50	6	..	1	41	2
Wood charcoal	79	2	..	11		8
Peat, dry.....	59	6	..	1.25	30	4
Straw.........	·36	5	..	0.5	38	4.5
Petroleum.....	85	13	..	..	2	..
Pennsylvania Cannel......	68.12	6.68	2.47	2.27	5.83	16 8
Indiana Can'l.	71.10	6.06	1.0	1.65	12.76	7.65

	Fixed Carbon.	Sulphur.	Volatile Matter.	Earthy Mat er.
Pennsylvania Semi-Bit'min's	73.11	.85	15.27	10.77
Semi-Anth'cite (Wilkesharre, Pa.)	88.90	..	7.68	3.49
True Anth'cite (Tamaqua, Pa.)	92.07	..	5.03	2.90

RATE OF COMBUSTION OF FUEL.

The rate of combustion of coal in steam boilers per square foot of fire grate per hour may be taken on the following basis:

Portable engine boilers..............................	9	to	16	pounds.
Vertical boilers.....................................	6	"	14	"
Cornish boilers......................................	12	"	15	"
Lancashire boilers	14	"	29	"
Marine boilers, natural draft........................	12	"	24	"
Marine boilers, forced draft.........................	20	"	34	"
Torpedo boat boilers.................................	40	"	70	"
Locomotive boilers...................................	40	"	100	"

WEIGHT AND COMPARATIVE FUEL VALUE OF WOOD.

One cord air-dried hickory or hard maple weighs about 4,500 pounds, and is equal to about 2,000 pounds coal.

One cord air-dried white oak weighs about 3,850 pounds, and is equal to about 1,715 pounds coal.

One cord air-dried beach, red oak or black oak weighs about 3,250 pounds, and is equal to about 1,450 pounds coal.

One cord air-dried poplar (whitewood), chestnut or elm, weighs about 2,350 pounds, and is equal to about 1,050 pounds coal.

One cord air-dried average pine weighs about 2,000 pounds, and is equal to about 925 pounds coal.

From the above it is safe to assume that two and one-fourth pounds of dry wood is equal to one pound average quality of soft coal, and that the full value of the same *weight* of different woods is very nearly the same—that is, a pound of hickory is worth no more for fuel than a pound of pine, assuming both to be dry. It is important that the wood be dry, as each 10 per cent of water or moisture in wood will detract about 12 per cent from its value as fuel.

EQUIVALENT MEASURES OF VOLUME.

1 imperial gallon = 277.274 cubic inches.
1 imperial gallon = 0.16045 cubic foot.
1 imperial gallon = 10 pounds.
1 United States gallon = 231.0 cubic inches = 8.34 pounds, nearly.
1 United States gallon = 0.8339 imperial gallon.
1 United States gallon = 3.8 liters of water.
A cubic foot of sea water = 64.00 pounds.
A cubic inch of sea water = 0.037037 pound.
A cubic foot of water = 62.32 pounds.
A cubic inch of water = 0.03616 pound.
A cylindrical foot of water = 48.96 pounds.
A cylindrical inch of water = 0.0284 pound.
A column of water 12 inches long, 1 inch square = 0.434 pound.
A column of water 12 inches long, 1 inch diameter = 0.340 pound.
The capacity of a 12-inch cube = 6.232 gallons.
The capacity of a 1-inch square 1 foot long = 0.0434 gallon.
The capacity of a 1-foot diameter 1 foot long = 4.896 gallons.
The capacity of a cylinder in gallons 1 yard long = 0.1 diameter squared.
The capacity of a 1-inch diameter 1 foot long = 0.034 gallon.
The capacity of a cylindrical inch = 0.002832 gallon.
The capacity of a cubic inch = 0.003606 gallon.
The capacity of a sphere 12 inches diameter = 3.263 gallons.
The capacity of a sphere 1 inch diameter = 0.00188 gallon.
1 imperial gallon = 1.2 United States gallon.
1 imperial gallon = 4.543 liters of water.
1 cubic foot of water = 6.232 imperial gallons.
1 cubic foot of water = 7.476 United States gallons.
1 cubic foot of water = 28.375 liters of water.
1 liter of water = 0.22 imperial gallon.
1 liter of water = 0.264 United States gallon.
1 liter of water = 61.0 cubic inches.
1 liter of water = 0.0353 cubic foot.

RELATIVE WEIGHTS OF METALS.

METALS.	Bar Iron.	Cast Iron.	Steel.	Brass	Copper.	Lead.	Zinc.
Bar Iron being 1.........	1.	0.93	1.01	1.09	1.15	1.48	0.92
Cast Iron " 1.........	1.07	1.	1.08	1.17	1.23	1.56	0.99
Steel " 1.........	0.99	0.92	1.	1.08	1.13	1.46	0.91
Brass " 1.........	0.92	0.85	0.93	1.	1.05	1.36	0.84
Copper " 1.........	0.87	0.81	0.88	0.94	1.	1.29	0.80
Lead " 1.........	0.68	0.63	0.69	0.74	0.78	1.	0.62
Zinc " 1.........	1.09	1.01	1.10	1.18	1.25	1.61	1.

APPENDIX II.

REFERENCES TO LITERATURE ON REFRIGERATION AND ALLIED SUBJECTS.

BOOKS, PAMPHLETS AND TREATISES.

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Arctic Machine Manufacturing Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Cleveland, Ohio.

Auldjo Machine Co. (Ice Making and Refrigerating Machine, Ammonia Compression System), Australia.

Austin Separator Co. (Oil Separators), Detroit, Mich.

Automatic Refrigerator Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Cleveland, Ohio, U. S. A.

Barber, A. H., Manufacturing Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Chicago, Ill.

Buffalo Refrigerating Machine Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Buffalo, N. Y.

Carbondale Machine Co. (Ice Making and Refrigerating Machinery, Ammonia Absorption System), Carbondale, Pa.

Case Refrigerating Machine Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Buffalo, N. Y.

Challoner's, Geo., Sons Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Oshkosh, Wis.

Clyde Engineering Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Sydney, Australia.

Cochran Co. (Ice Making and Refrigerating Machinery, Carbonic Anhydride System), Lorain, Ohio.

Creamery Package Manufacturing Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Chicago, Ill.

De La Vergne Refrigerating Machine Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), New York, N. Y.

Direct Separator Co. (Water and Oil Separators), Syracuse, N. Y.

Farrell & Rempe Co. (Wrought Iron Coils and Ammonia Fittings), Chicago, Ill.

Frick Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System, and Corliss Engines), Waynesboro, Pa.

Garlock Packing Co. (Ammonia Packings), Palmyra, N. Y.

Gifford Bros. (Ice Elevating, Conveying and Lowering Machinery), Hudson, N. Y.

Gloekler, Bernard (Cold Storage Doors and Fasteners), Pittsburg, Pa.

Goodsell Packing Co. (Ammonia Packing), Chicago, Ill.

Hall, J. & E., Ltd. (Ice Making and Refrigerating Machinery, Carbonic Anhydride System), London, E. C., England.

Harrisburg Pipe and Pipe Bending Co., Ltd. (Coils and Bends, and Ammonia Fittings and Feed Water Heaters), Harrisburg, Pa.

Haslam Foundry and Engineering Co. (Ice Making and Refrigerating Machinery, Ammonia Absorption System), Derby, England.

Henderson, Thoens & Gerdes (Ice Making and Refrigerating Machinery, Ammonia Absorption System) New Orleans, La.

Hercules Ice Making and Refrigerating Machinery (Ammonia Compression System), Sydney, Australia, and Chicago, Ill.

Hohmann & Maurer Mfg. Co. (Thermometers), Rochester, N. Y.

Hoppes Manufacturing Co. (Water Purifiers and Heaters), Springfield, Ohio.

Hoppes Manufacturing Co. (Steam Separators and Oil Eliminators), Springfield, Ohio.

Humble & Nicholson (Ice Making and Refrigerating Machinery), Geelong, Australia.

Ideal Refrigerating and Manufacturing Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Chicago, Ill.

Jarecki Manufacturing Co. (Ammonia and Steam Fittings), Erie, Pa.

Kilbourn Refrigerating Machine Co., Ltd. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Liverpool, England.

Kroeschell Bros. Ice Machine Co. (Ice Making and Refrigerating Machinery, Carbonic Anhydride System), Chicago, Ill.

Lewis Manufacturing Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), New York.

MacDonald, C. A. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Chicago, Ill., and Sydney, N. S. W.

Nason Mfg. Co. (Ammonia and Steam Fittings), New York, N. Y.

Newburgh Ice Machine and Engine Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Newburgh, N. Y.

New York Fastener Co. (Door Fasteners), Newark, N. J.

Pennsylvania Iron Works Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Philadelphia, Pa.

Philadelphia Pipe Bending Works (Wrought Iron Coils and Bends), Philadelphia, Pa.

Pontifex & Sons (Ice Making and Refrigerating Machinery, Ammonia Absorption System), London.

Pulsometer Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), London.

Remington Machine Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Wilmington, Del.

Ruemmeli & Siebert Refrigerating Machine Co. (Ice Making and Refrigerating Machinery, Gradirworks, Fittings, etc.), St. Louis, Mo.

Schwabe, J. S., & Sohn (Ice Making and Refrigerating Machinery, Ammonia Compression System), Saxony, Germany.

Seyboth, L. (Ice Making and Refrigerating Machinery), Munich, Germany.

Siddely & Co. (Ice Making and Refrigerating Machinery, Ammonia Absorption System), Liverpool, England.

Siebe & Sorman (Ice Making and Refrigerating Machinery, Ether System), London.

Sterne & Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), London, England.

Stevenson Co., Limited (Cold Storage Doors), Chester, Pa.

Tight Joint Co. (Ammonia Fittings), New York City, N. Y.

Triumph Ice Machine Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Cincinnati, Ohio.

Vaass & Littmann (Ice Making and Refrigerating Machinery (both Absorption and Compression Systems), Halle-a-Salle, Germany.

Vilter Mfg. Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System, and Corliss Engines), Milwaukee, Wis.

Vogt, Henry, Machine Co. (Ice Making and Refrigerating Machinery, Ammonia Absorption System), Louisville, Ky.

Vulcan Iron Works (Ice Making and Refrigerating Machinery, Ammonia Compression System), San Francisco, Cal.

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Wolf Co., Fred W. (Ice Making and Refrigerating Machinery, Ammonia Compression System), Chicago.

Wolf Co., Fred W. (Ammonia Fittings and Ice and Refrigerating Machinery Supplies), Chicago.

Wood, Wm. T. & Co. (Ice Tools), Arlington, Mass., U. S. A.

York Manufacturing Co. (Ice Making and Refrigerating Machinery, Ammonia Compression System), York, Pa., U. S. A.

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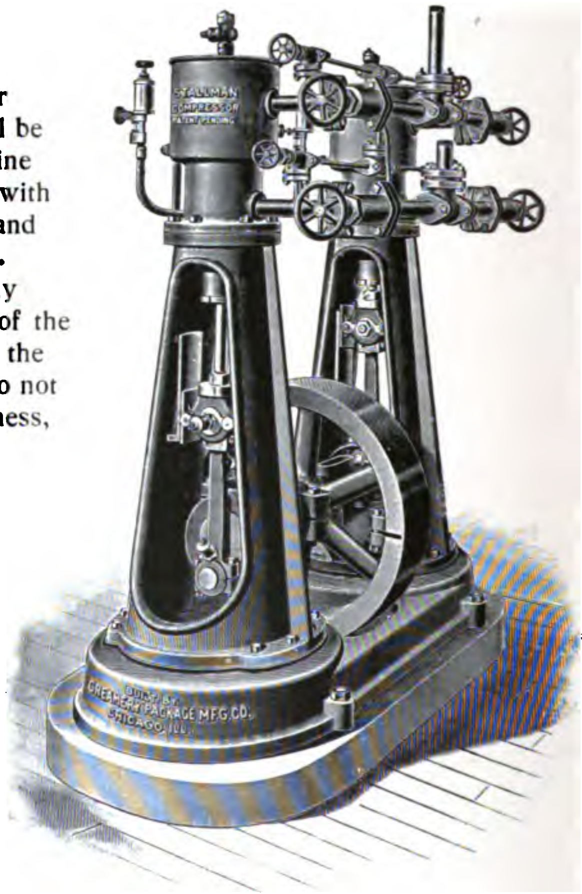

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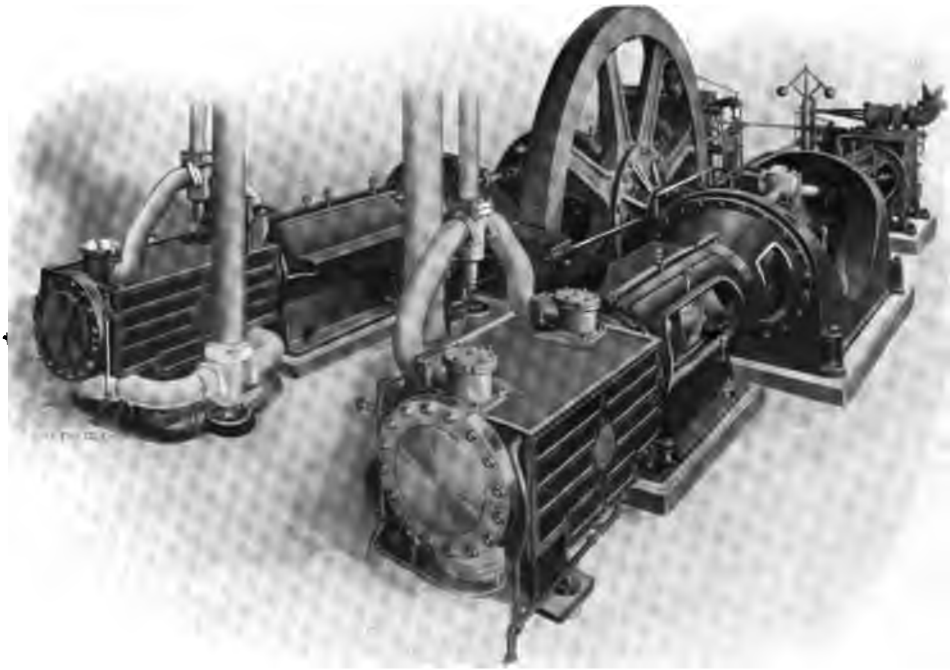

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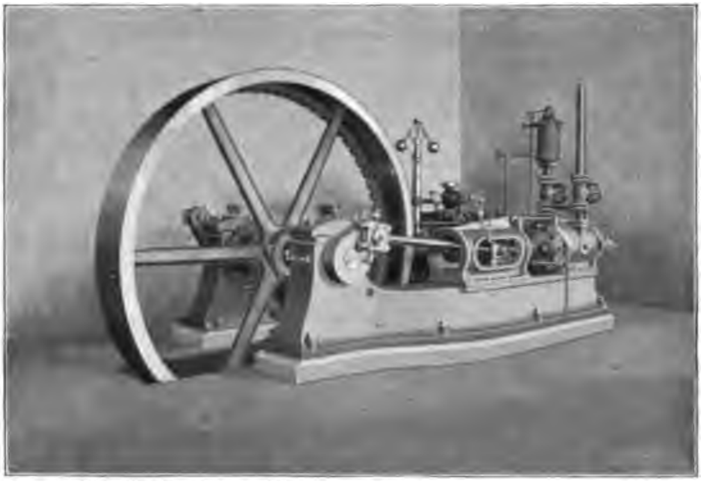

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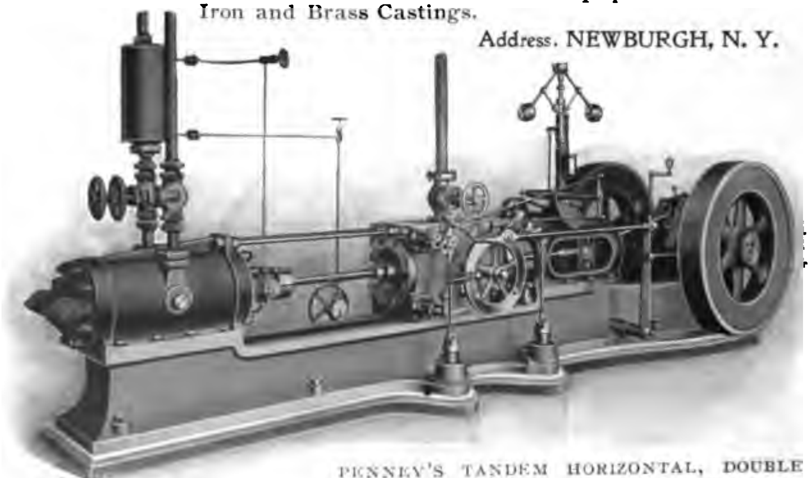

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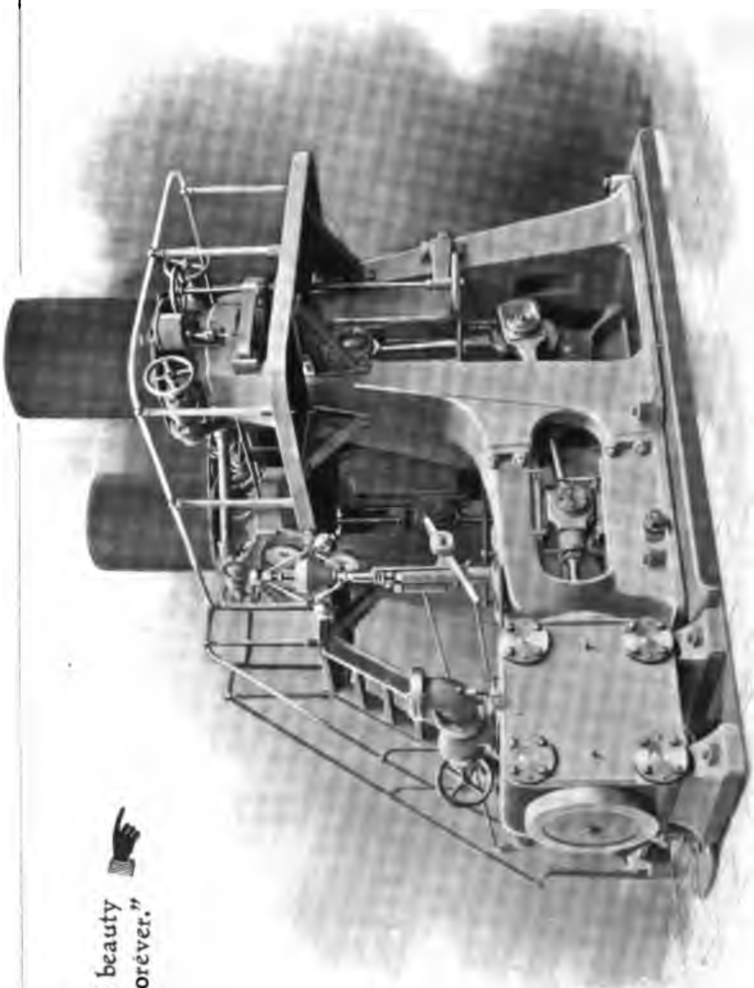

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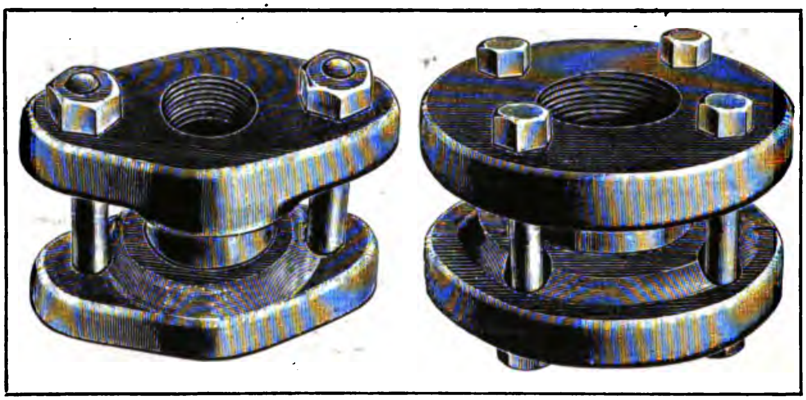

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